# STEAM-TURBINES

BY

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# PREFACE.

In writing this book I have aimed to give in logical order the fundamental principles of steam-turbine design, with examples of their application, and to show the results obtained in engineering practice.

The development of the steam-turbine has been so rapid that many of the problems involved, while solved more or less satisfactorily for constructive purposes, have not been put upon a scientific basis. Foremost among these problems is that of the velocity of steam-flow under given conditions,—important not only for an understanding of the operation of the turbine, but for predicting the results to be expected from a given set of conditions. My principal incentive has been the desire to analyze and correlate the results of certain important experimental investigations, and to show how these results could be used in connection with the well-known laws of hydraulics and thermodynamics as applied to steam-turbines. In stating these laws I have attempted to develop the expressions in a simple and direct manner, and to give numerical and graphical solutions illustrating the principles involved.

The book is not intended to be or to take the place of a treatise on either hydraulics or thermodynamics, but it has seemed best to give in outline the development of such parts of those subjects as are most necessary for acquiring the working knowledge which it is the object of the book to impart. I have

attempted, therefore, to discriminate between essential principles and such discussions as are chiefly of scholarly interest.

A large part of the experimental data used in the book was obtained by Professor Gutermuth, of Darmstadt, Dr. Stodola, of Zürich, Mr. George Wilson, of Manchester, Mr. Walter Rosenhain, of Cambridge, and Professor Rateau, of Paris. I have taken the material from various sources, and have endeavored to give credit in all cases. The work on nozzles and buckets combined was done in the Sibley College laboratories, and a series of similar experiments is now in progress there, in which the exhaust is led into a condenser maintaining such vacuum conditions as are used in practice.

I am especially indebted to the officials of The General Electric Company, The Westinghouse Machine Company, The Allis-Chalmers Company, and The De Laval Steam Turbine Company, for opportunities for taking extended observations at their works, and for permission to use data and material for illustrations. Especial thanks are also due to Professor R. C. Carpenter for placing at my disposal valuable experimentally obtained data; and to Messrs. A. G. Christie, C. E. Burgoon, and J. C. Wilson for assistance in making calculations and plotting curves.

C. C. T.

ITHACA, N. Y., January, 1906.

# PREFACE TO THE THIRD EDITION.

In presenting this third edition the writer wishes to call attention to the new problems in the design of the Curtis and the Parsons types of turbine, to the suggestions regarding turbine analysis, and to the Diagram of Heat-contents of Steam, the superheated region of which is plotted from the results of the writer's recent investigation of the specific heat of superheated steam.* This diagram is laid out as suggested by Dr. Mollier, the co-ordinates being Entropy, and Total Heat-contents, and is exceedingly convenient because heat-units are read on straight lines instead of on curves as in the Temperature-entropy Diagram. The present edition contains also new material relating to the application of steam-turbines to marine propulsion, including illustrations of some of the most recent turbine steamers.

The object of the book is, as before, to set forth the principles essential to those who wish to equip themselves for taking up steam-turbine work. Only such details relating to present practice in turbine construction have been given, therefore, as would suffice to illustrate the application of the principles.

C. C. T.

Ithaca, N. Y., November, 1907.

* American Society of Mechanical Engineers, December, 1907.

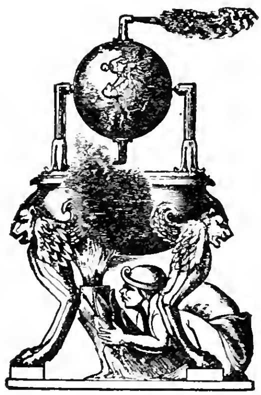
# CONTENTS.

# INTRODUCTION.

Rotation in a steam-turbine is caused by particles of steam acting upon suitably formed surfaces attached to the rotating part of the machine. Steam consists of very small particles or molecules possessing mass, and the heat in steam may be caused to impart high velocity to its own particles. This is accomplished by allowing the steam to fall suddenly in temperature and thus to give up its heat as work in expanding its volume and expelling its own substance from a place of higher to one of lower pressure. If the expansion takes place in a given direction, as when steam flows from a nozzle, the action is somewhat similar to that occurring in the barrel of a gun when the charge of powder burns, forming a gas of high temperature which quickly expands, driving before it the projectile and also the particles of gas and burnt powder. The substance expelled from the gun, having had work done upon it, attains a certain velocity and is capable of giving up its energy, minus certain losses, to whatever objects may be in the way tending to retard or change the motion of the mass.

When a substance, such as steam or water or gas, flows through a nozzle and has its motion accelerated during the flow, a reaction occurs opposite in direction to the flow and tending to move the nozzle. The recoil of a gun or of a hose-nozzle is an example of such a reaction. In turbines of the so-called reaction type this phenomenon is utilized for pro-

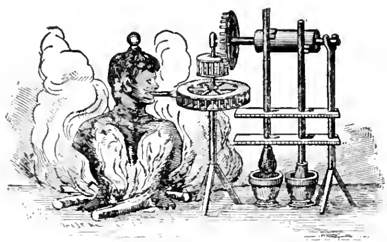
ducing motion of the rotating part. A true reaction-turbine may be compared to a pinwheel in the periphery of which small charges of powder are exploded and from which the resulting gases are expelled in such a direction as to give the wheel a motion of rotation due to the reaction accompanying the expulsion of the charge. The energy possessed by the charge leaving the pinwheel might be directed upon another movable wheel, and the latter be rotated by the impulse thus received. Such a combination of reaction and impulse takes place in



Hero's reaction-turbine.

what is called the reaction-turbine. The operation is as follows: The stationary casing of the machine holds a row of guide-blades in front of each row of moving blades. The space between each two guide-blades forms a nozzle through which the steam passes on its way to the moving blades. The pressure between the guide-blades and the moving blades is less than that in the space before the guide-blades; therefore the steam expands as it passes through the guide-blades, and its motion is accelerated as the pressure falls during the expansion. The steam strikes the moving blades with the velocity it has upon leaving the guide-blades, and exerts an impulse as the moving blades change the velocity of the steam. But there is a still lower pressure beyond the moving blades than before them, and therefore the steam expands still further in

the moving blades and accelerates the velocity of its own particles according to the amount of heat given up during the fall of pressure accompanying the expansion. The moving blades discharge the steam in a direction opposed to that of their rotation, and the reaction accompanying the acceleration of the steam in the moving blades acts to produce rotation, just as did the impulse when the steam first struck the moving blades. The rotative effect is thus produced by both impulse and reaction, and the name "reaction-turbine" should in this case give place to "impulse-and-reaction turbine."



Branca's impulse-turbine.

In an impulse-turbine nozzles are held in the frame of the machine, at rest relatively to the earth, and steam expands in the nozzles, giving up its heat to an extent depending upon the degree of expansion, and to that extent does work upon its own mass, discharging it upon the movable part of the machine. The latter absorbs energy from the rapidly moving particles of steam, and gives out the energy, minus certain losses, as rotative effort. The steam particles receive in the nozzles all the mechanical energy they are to possess, for there is in the ideal, single-stage impulse-turbine no fall in pressure after the steam leaves the nozzles. There is therefore the same pressure on the two sides of the rotating row of blades, and the latter simply receive an impulse due to the reduction in kinetic energy which the steam experiences during its passage through the blades.

In the many-stage impulse-turbine the fall in pressure and

temperature occurring in any one stage is limited according to the work that is desired to be produced by a single stage. Thus the steam still possesses energy after its passage through the blades in a given stage, and this remaining energy may be used in a succeeding stage in the manner described. The smaller the amount of energy remaining in the steam after passage through the final stage of the turbine, the more efficient is the machine as a heat-engine.

In general, steam-turbine design is concerned primarily with the use of the energy of rapidly moving masses of steam and with the heat transformations which give rise to the motion of the steam. A knowledge of the principles underlying these phenomena is therefore necessary, and the first three chapters were written to make the fundamentals clear. In Chapters IV, V, and VI, the flow of steam through orifices and nozzles is discussed, and experimentally obtained results are given in order to connect what would be expected to occur under ideal conditions with what actually occurs in engineering practice.

In the remaining chapters the principles of turbine design and operation are discussed, and it has been the constant aim in this work to show in what way the results to be expected may be predicted by the proper use of experimental data.

# CLASSIFICATION OF STEAM-TURBINES.

1. Impulse turbines. Equal pressure on the two sides of any row of buckets.

2. Reaction, or Impulse-and-reaction turbines. — Fall of pressure in passing any row of buckets.

**Impulse type. Partial peripheral admission, excepting Hamilton-Holzwarth.**

- Nozzles inclined to Plane of Rotation.
  - (*a*) Single stage, consisting of one set of nozzles and one row, or wheel, of buckets. (Example, De Laval turbine.)
  - (*b*) Velocity compounded, single stage, one set of nozzles and several rows of moving buckets, with intermediate guides. (Curtis.)
  - (*c*) Pressure Compounded, several compartments, or stages, each containing one set of nozzles and one set of moving buckets. (Rateau, Zoelly, Hamilton-Holzwarth.)
  - (*d*) Several stages; both pressure and velocity compounded. Each compartment, or stage, contains one set, (perhaps divided into two groups) of nozzles, and two or more rows of moving buckets, and one or more rows of stationary buckets. (Curtis, vertical and horizontal.)
- (*e*) One or more stages. Buckets of Pelton type cut in rim of wheel. Nozzles in plane of rotation. (Riedler-Stumpf.)

**Full peripheral admission.** — Many stage turbine, or Parsons type. Steam acts by both impulse and reaction.

**Partial peripheral admission in Impulse stages, and full peripheral admission in Parsons stages.** — Combinaton of Impulse stages with those of the Impulse-and-reaction type. Generally one or more Impulse stages at high pressure end of turbine, followed by a large number of Impulse-and-reaction, or Parsons stages.

# STEAM-TURBINES.

## CHAPTER I.

### GENERAL PRINCIPLES RELATING TO THE ACTION OF STEAM UPON TURBINE-BUCKETS.

THE effect of steam striking against and leaving the moving parts of a turbine may be analyzed by means of the principles discussed in the present chapter.

A force acting upon a body tends to change the position of the body. If the latter is at rest relatively to the earth, it is said to have zero velocity, and a force may act so as to impart to the body a certain motion. If the body is in motion before the force acts upon it, the effect of the force is to increase or decrease the rate of motion of the body, or else to change its direction of motion. Or, the force may change both the rate and the direction of motion. Change of rate of motion is called *acceleration*. If a force increases the velocity of a body, it is said to produce a *positive acceleration*. If the effect of the force is to reduce the velocity, it is said to produce a *negative acceleration*.

If the mass of a body be known, and the acceleration in a given direction due to a force be also known, the magnitude of the force can be calculated. It follows, therefore, that a force can be measured by the acceleration it produces when it acts upon definite quantities of matter whose conditions of

motion are known. If a force communicates equal increments of velocity in equal lengths of time, it is said to be a *uniform force.*

If a force acts upon a body in a fixed direction, and produces an acceleration $f$,—that is, if it adds $f$ units of velocity per unit of time,—then in $t$ units of time the velocity generated is $V=ft$.

The space passed over in the time $t$ is the product of the mean velocity $\frac{V}{2}$ and the time $t$.

If space passed over is $s$, then

$$s=\frac{ft}{2}\times t=\tfrac{1}{2}ft^2.$$

But $t=\frac{V}{f}$, and therefore $\quad s=\tfrac{1}{2}f\times\frac{V^2}{f^2}=\frac{V^2}{2f}$.

This may be written $\quad V^2=2fs.$

Applying this general statement to the case of a body falling freely towards the earth, under the influence of the force of gravitation, whose acceleration is called $g$, the space through which the body must fall in order that it may attain the velocity $V$, is $h=\frac{V^2}{2g}$.

If a free body of mass $M$ is acted upon by a force $F$, in a fixed direction during a given time, a certain acceleration of the motion of the body will take place. If the force $F$ acts upon a mass of $2M$ during the same length of time, the acceleration, or increase of velocity, will be only half as great as in the first instance. To produce the same effect in the same time upon $2M$ as was produced by $F$ upon $M$, the force must be $2F$.

Further, if a force $F$ produces an increase of velocity, $V$, in a mass $M$ in a given time, it will require a force of $2F$ to produce a velocity of $2V$ in the same mass in the same time.

And if a certain force imparts in one second to a mass weighing 2 pounds a velocity of 2 feet per second, it is capable of imparting to a mass of 4 pounds a velocity of only 1 foot per second.

From these facts it is seen that the force required to change the motion of matter varies as the acceleration, or velocity acquired in a given time, and as the mass acted upon. It therefore varies as their product, and since a force $F$, which accelerates the velocity of a mass $M$ by an amount $j$ per unit of time, varies as the product $Mj$, the equation may be written $F=CMj$, where $C$ is some constant.

The unit of mass, as used in engineering, is a derived unit, and its value may be found in terms of force and acceleration by letting $C=1$. The earth attracts every mass of matter upon its surface with a force (called the force of gravitation) capable of imparting to the mass an acceleration of about 32.2 feet per second per second. The magnitude of the force is proportional to the amount of matter, or the mass, acted upon, and is called the *weight* of the mass. The weight of a certain mass of platinum has been accepted as the unit force, and is called the *pound*. If $F=1$ pound and $j=32.2$ feet per second per second, the equation may be written:

$$\frac{1}{32.2}=M=\text{the amount of mass in 1 pound weight.}$$

The value of $M$ in this equation can be made equal to unity only by multiplying the left-hand member by 32.2, and therefore the unit mass is so much mass as weighs 32.2 lbs. To express quantities of mass, then, in terms of weight, it is necessary to divide the weight of the mass by 32.2, or $M=W\div 32.2$. Calling the acceleration due to gravity $g$, the equation becomes

$$M=\frac{W}{g}, \quad \text{or} \quad W=Mg.$$

The equation expressing the relation between force, mass, and acceleration is, then,

$$F=Mj=\frac{W}{g}j,$$

where $F$ is the force which produces in the mass $\frac{W}{g}$ the acceleration $f$.

A weight $W$, if allowed to fall, is accelerated by an amount $g$ ft. per second. Forces are proportional to the acceleration they produce upon bodies free to move, and, therefore, any force $F$ which can produce an acceleration $f$ is related to $W$ and $g$ by the equation $\frac{F}{f}=\frac{W}{g}$. Hence the force $F$ which can give a velocity of $f$ ft. per second to a mass $W$, in 1 second, is equal to $\frac{Wf}{g}=Mf$, where $M=$the mass accelerated.

If a stream of any substance, such as water, gas, or steam, or of a mixture of steam and water, moves with a velocity $V$, in a fixed direction, then if $W$ is the weight of the substance passing a given cross-section of the conducting channel per second, the work it is capable of doing, or the energy it possesses by reason of its mass and velocity, is the same as the energy developed by a body falling freely under the action of gravity through a height $h$, and thereby acquiring the velocity $V$.

If $K$ be the kinetic energy of the stream, or its capacity to do work, then

$$K=Wh=\frac{WV^2}{2g}. \qquad (2)$$

Hence the energy of a stream of constant cross-section is proportional to the square of its velocity.

If a nozzle delivers $W$ pounds of the substance per second with a uniform velocity $V$, it may be considered that a constant impulsive force $F$ has acted upon the weight $W$ for one second and then ceased. During this second the substance has changed its velocity from $O$ to $V$, and has traversed the space $\frac{1}{2}V$. Therefore the work $F\times\frac{V}{2}$ has been done upon the substance by the impulsive force $F$.

The energy of the jet is $\frac{WV^2}{2g}$, and this must equal the work which has been done upon the jet, or $F\times\frac{V}{2}$

Hence $$F \times \frac{V}{2} = \frac{WV^2}{2g}, \quad \text{or} \quad F = \frac{WV}{g}. \qquad . \quad . \quad . \quad . \quad . \quad (3)$$

If $A$ = the area of cross-section of the jet, and the weight of the substance per cubic unit $= w$, then $W = wAV$, or

$$F = \frac{wAV^2}{g}.$$

The jet is capable of exerting an impulse equal to $F$ upon any object in its way, and therefore the impulse of a jet of constant cross-section varies as the square of its velocity.

The force $F$ acts for one second upon each $W$ pounds of substance which pass a given section. But as there is only the amount $W$ passing per second, the force $F$ is continuously exerted and becomes a continuous impulsive pressure.

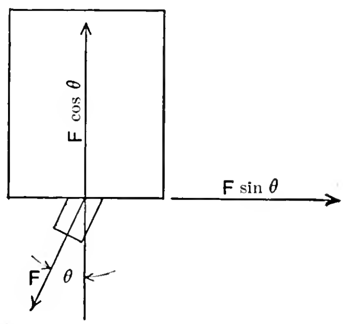



FIG. 1.

A stream flowing from an orifice produces a reaction equal in value, and opposite in direction, to the impulse the stream is capable of producing upon an object against which it may strike. In the direction of the jet the impulse produces motion. In the opposite direction it produces a pressure tending to move the orifice or nozzle and whatever is rigidly connected therewith.

The force $F = \frac{WV}{g} = \frac{wAV^2}{g}$, is exerted in the line of action of

the jet, and its force in any other direction is the component of the force $F$ in that direction.

If steam, for example, issues vertically downward from an orifice in the base of a vessel, it exerts an upward reaction $F$ and a horizontal reaction 0. If its direction of issue is inclined 20° to the vertical, its upward reaction is $F\cos 20°$, and its horizontal reaction is $F\sin 20°$. (Fig 1.)

If a stream moving with velocity $V_1$ is retarded so that its velocity becomes $V_2$, its impulse at first is $W\dfrac{V_1}{g}$ and after retardation $W\dfrac{V_2}{g}$. The dynamic pressure developed is

$$P=\frac{W(V_1-V_2)}{g}.$$

It is by means of the pressure resulting from change of velocity or of direction of flow, or both, that turbine-wheels transform the energy of moving water, steam, or gas into useful work.

*Example 1.*—200 pounds of water flows each second from an orifice having a cross-sectional area of .064 sq. ft. What is the velocity of flow?

Quantity = area × velocity, or

$$200\div 62.4=3.2 \text{ cu. ft. per second.}$$
$$3.2\div 0.064=50 \text{ ft. per second.}$$

What is the horse-power of the jet?

Energy, or capacity to do work,

$$=\frac{WV^2}{2g}=\frac{200.\times(50.)^2}{64.4}=7760. \text{ ft.-pds. per second.}$$
$$7760.\div 550.=14.1 \text{ horse-power.}$$

What is the reaction against the vessel from which the water flows?

$$\text{Reaction}=\text{impulse}=F=\frac{WV}{g}=\frac{200\times 50}{32.2}=311 \text{ pounds.}$$

If the water should act upon a revolving wheel, leaving the buckets at a velocity of 30 ft. per second, what horse-power would be given up to the wheel? Neglect losses.

$$\text{Energy given up} = \frac{W(V_1^2 - V_2^2)}{2g} = \frac{200((50)^2 - (30)^2)}{64.4}$$

$$= 4960 \text{ ft.-pds. per second.}$$

$$4960 \div 550 = 9.04 \text{ horse-power.}$$

Efficiency of wheel, disregarding friction, $= 9.04 \div 14.1 = .64$.

If the water at 30 ft. per second should be used to drive another wheel, leaving its buckets at a velocity of 10 ft. per second, what would be the efficiency of the two wheels combined?

$$\text{Horse-power of second wheel} = \frac{200 \times ((30)^2 - (10)^2)}{64.4 \times 550} = \phantom{1}4.52$$

$$\text{“} \qquad \text{“ first} \qquad \text{“} \qquad = \phantom{1}9.04$$

$$\text{“} \qquad \text{“ two wheels} \qquad = 13.56$$

$$\text{Efficiency} = 13.56 \div 14.1 = .96 + .$$

The same total efficiency would of course be obtained by using the first single wheel, if the water should leave it at the velocity of 10 ft. per second.

$$\text{Thus,} \quad \frac{200((50)^2 - (10)^2)}{64.4 \times 550} = 13.5 + \text{horse-power.}$$

$$13.5 \div 14.1 = .96, \text{ approximately.}$$

The efficiency of the system of wheels is evidently

$$\frac{V_1^2 - V_2^2}{V_1^2} = \frac{2500 - 100}{2500} = .96.$$

*Example 2.*—Suppose 100 pds. steam to flow per second from the orifice of the previous example, what would be the horse-power of the jet?

The area of the orifice is .064 sq. ft. (about 3.4 ins. diam.). Let the volume of steam per pound=2.5 cu. ft. in the orifice.

$$\frac{100\times 2.5}{0.064}=3900 \text{ ft. per second velocity.}$$

Energy, or capacity for doing work, $=\frac{WV^2}{2g}=\frac{100\times(3900)^2}{64.4}=$

23,600,000 ft.-pds. per second.

$$\frac{23,600,000}{550}=42,900 \text{ horse-power.}$$

If the steam in such a jet should all be used upon a turbine, leaving same at a velocity of 1000 ft. per second, what horse-power would be developed, disregarding frictional and thermal losses?

$$\text{Energy given up}=\frac{W(V_1^2-V_2^2)}{2g}=\frac{100((3900)^2-(1000)^2)}{64.4}$$

$$=22,200,000 \text{ ft.-pds. per second.}$$

$$\frac{22,200,000}{550}=40,400 \text{ horse-power.}$$

$$\text{Efficiency}=\frac{40,400}{42,900}=.94.$$

What would be the reaction against a steam-nozzle from which such a stream was emitted?

$$F=\frac{WV}{g}=\frac{100\times 3900}{32.2}=12,100 \text{ pounds.}$$

*Example 3.*—If a jet has a cross-sectional area of 1 sq. inch, how many cubic feet of air at atmospheric pressure must it emit per second in order that its impulse may be 200 pounds?

1 cu. ft. of air at atmospheric pressure and 60 degrees F. weighs approximately 1/13 pound.

If $w$ = weight of air per cu. ft. and $A$ = area of orifice in sq. ft., then

$$F=\frac{WV}{g}=\frac{2wAV^2}{2g}=\frac{wAV^2}{g}=200 \text{ pounds.}$$

$$\frac{1}{13}\times\frac{1}{144}\times\frac{V^2}{32.2}=200, \quad \text{or}$$

$$V=\sqrt{200.\times32.2\times144.\times13.}=3490. \text{ ft. per second.}$$

$$\frac{3490.}{144.}=24. \text{ cu. ft. per second.}$$

*Example 4.*—If a tube $T$ is 1″ dia. and delivers 0.3 cu. ft. of water per sec. compute the dynamic pres. against the plane.

$$A=\frac{.785}{144} \text{ sq. ft.} \quad W=.3 \text{ cu. ft.}=18.7 \text{ pds.}$$

$$V=\frac{.3\times144}{.785}=55 \text{ ft. per sec.}$$

$$F=\frac{WV}{g}=\frac{18.7\times55}{32.2}=32 \text{ pds., approx.}$$

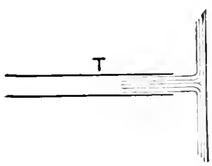



Fig. 2.

*Example 5.*—If a nozzle having a cross-sectional area of 0.1 sq. in. discharges 500 pounds of steam per hour, and experiences a reaction against itself of 15 pounds, what is the velocity of the issuing jet of steam?

Since the reaction is equal to the impulse the jet is capable of exerting, it equals

$$R=\frac{WV}{g}, \quad \text{or} \quad V=\frac{Rg}{W}=\frac{15\times 32.2}{\frac{500}{3600}}=3480 \text{ ft. per sec.}$$

**Action of Fluid upon Vanes.**—Let a stream of fluid enter a stationary vane tangentially to the surface at $A$, Fig. 3, and let

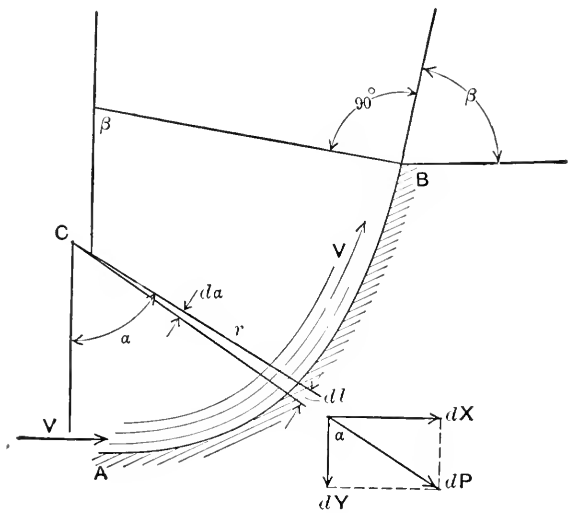



FIG. 3.

it traverse the vane to $B$ with the velocity it had at $A$. This condition would be possible if the fluid experienced no frictional resistance to its passage along the surface. As the fluid enters the vane its tendency is to continue flowing in the direction it has at $A$, but it is prevented from maintaining this direction of flow by the curvature of the surface it has to traverse. The vane has to oppose a resistance to the tendency of the fluid to flow in its original direction, in order to effect the change in direction, and that resistance amounts to a force pushing the fluid towards the center of curvature of the vane at each point

of the path. The force causing the stream to take the direction of the vane surface is similar to the pull on a string by which a weight is held and caused to swing about the point at which the string is held. If a certain weight of fluid, for the instant in which it covers the distance $dl$, is rotating about a center at $C$, it is exerting a pressure in a direction normal to the surface at $dl$, and that pressure is equal in amount to the centrifugal force exerted by a body having the same weight as the water on $dl$, and moving with the velocity $V$ at a distance $r$ from the center of rotation. The centrifugal force $=\frac{WV^2}{gr}$, or, if the area of cross-section of the stream is $A$ sq. ft. and the fluid weighs $w$ pounds per cubic foot, the weight $W = Awdl$ and the centrifugal force on $dl$ is

$$dP = \frac{Adlw}{g} \times \frac{V^2}{r}.$$

The pressure on the small area of length $dl$ in the direction which the stream had when it entered the vane is

$$dX = dP \sin \alpha,$$

and in the direction perpendicular to that of the stream at entrance it is $dY = dP \cos \alpha$.

The total angle subtended by the surface of the vane is $\beta$, and upon each elementary area of width $dl$ there is the force $dP$ pressing against the vane. The total component of the force in the direction of $dX$ is

$$P_X = \int_{\alpha=0}^{\alpha=\beta} dX = \int_{\alpha=0}^{\alpha=\beta} dP \sin \alpha$$

$$= \frac{AwV^2}{g} \int_0^\beta \frac{dl \sin \alpha}{r}.$$

But $dl = rd\alpha$, and therefore

$$P_X = \frac{AwV^2}{g} \int_0^\beta \sin \alpha d\alpha$$

$$= \frac{AwV^2}{g}(1 - \cos \beta).$$

Similarly, $$P_Y = \int_0^\beta dP \cos \alpha = \frac{AwV^2}{g} \sin \beta.$$

The resultant impulse on the vane is

$$P_R = \sqrt{P_X^2 + P_Y^2} = \frac{AwV^2}{g}\sqrt{2(1 - \cos \beta)}.$$

Since the volume of fluid passing the surface per second is equal to the cross-sectional area of the stream multiplied by the velocity, and since the volume multiplied by the weight per cubic unit equals the total weight flowing per second,

$$\text{Weight flowing per second} = W = wAV.$$

The expressions for impulse may then be written as follows:

$$P_X = \frac{WV}{g}(1 - \cos \beta); \quad \ldots \ldots \quad (4)$$

$$P_Y = \frac{WV}{g} \sin \beta; \quad \ldots \ldots \quad (5)$$

$$P_R = \frac{WV}{g}\sqrt{2(1 - \cos \beta)}. \quad \ldots \ldots \quad (6)$$

The direction of $P_R$ with respect to $P_X$ and $P_Y$ is given by the equation

$$\cot \alpha = \frac{P_Y}{P_X} = \frac{\sin \beta}{1 - \cos \beta}.$$

The matter may be approached by a method more direct, though less satisfactory from an analytical standpoint, as follows: If a stream of constant cross-sectional area flows with a constant velocity $V$ and is deflected by the surface of a vane, as in Fig. 4, the impulse it is capable of producing in the direction of flow is the same at all points of the path. The reaction exerted by the stream in the direction opposite to that of flow is also constant. As the stream enters the

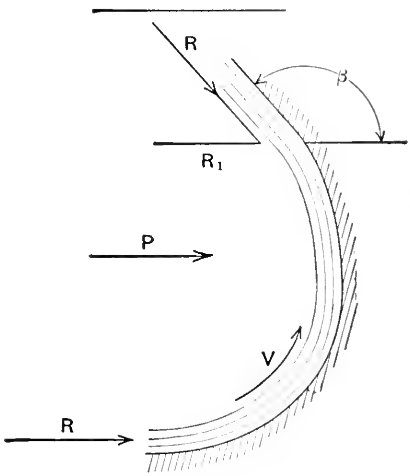



FIG. 4.

surface it exerts its impulse $R$ in the direction of flow, and as it leaves the surface the reaction $R$ is exerted in a direction opposite to that of flow.

Let $P$ be the dynamic pressure, or the impulse produced in the direction of the initial motion as the jet strikes the vane, and let $R_1$ be the component in that direction of the reaction of the jet as it leaves the vane. Then, if $\beta$ is greater than 90°, as shown in Fig. 4, the total pressure upon the vane is

$$P = R + R_1 = R + R \cos (180° - \beta) = R(1 - \cos \beta).$$

If $\beta$ is less than 90°,

$$P = R - R_1 = R - R \cos \beta = R(1 - \cos \beta).$$

The result is the same in the two cases, and the value of the impulse is seen to depend upon the angle of exit of the vane. Since the impulse $R=\frac{WV}{g}$, the total pressure is, as before found,

$$P=\frac{WV}{g}(1-\cos\beta).$$

If $\beta=0$, as when a stream flows along a straight surface, $P=0$.

If $\beta=90°$, as in Fig. 5, $\cos\beta=0$ and $P=\frac{WV}{g}$.

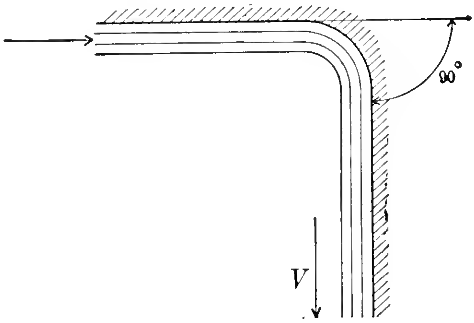



Fig. 5.

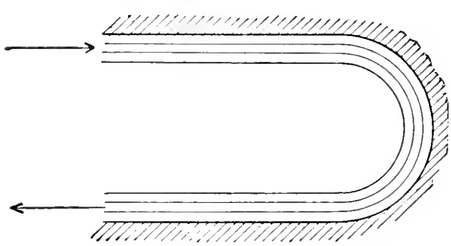


Fig. 6.

If $\beta=180°$, as in Fig. 6, a complete reversal of direction occurs, and

$$P=2R=2\frac{WV}{g}.$$

If the direction in which it is required to find the dynamic pressure makes an angle $\alpha$ with the direction of the entering jet, and an angle $\beta$ with that of the jet when it leaves the vane,

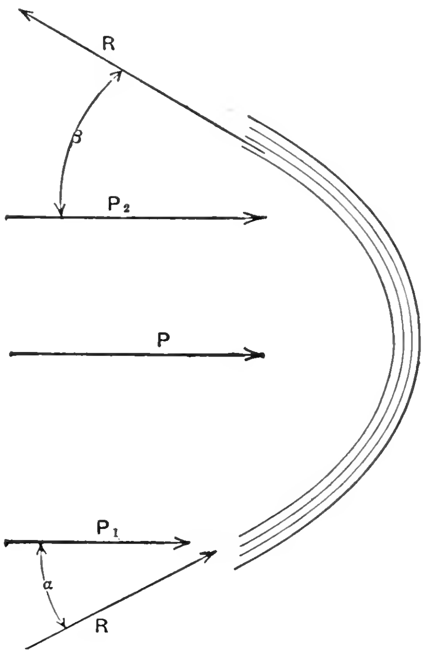



FIG. 7.

the components of the impulsive pressure in the direction of $P_1$ and $P_2$, Fig. 7, are

$$P_1 = R \cos \alpha,$$

$$P_2 = R \cos \beta.$$

$$P = R(\cos \alpha + \cos \beta) = \frac{WV}{g}(\cos \alpha + \cos \beta).$$

If $\alpha = 0°$ and $\beta = 90°$, as in Fig. 5, then $P = R$.

If $\alpha = 0$ and $\beta = 0$, as in Fig. 6, then $P = 2R$.

**Let a vane, or "bucket," move with velocity u,** in a straight line, when acted upon by a jet of fluid having a velocity $V$ in the same direction as the motion of the vane.

Let the stream at exit from the vane have a direction making an angle $\beta$ with a line drawn in direction opposite to that of the velocity $u$. The velocity of the jet relatively to the vane is $V-u$, and a dynamic pressure is produced upon the vane in the direction of motion, just as if the vane were at rest

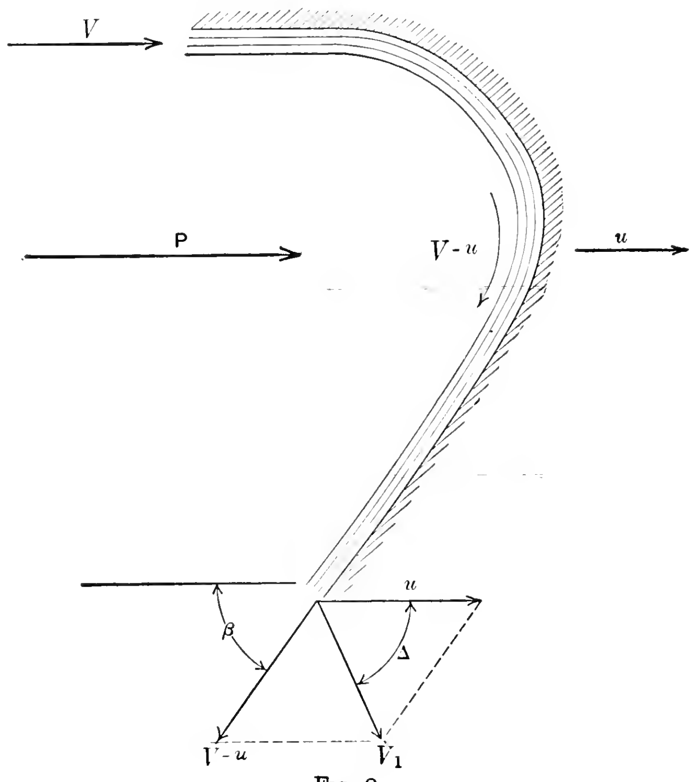



FIG. 8.

and were acted upon by a jet moving with the absolute velocity $V-u$.

For a surface at rest the action of a jet having a velocity $V$ produces a pressure in the direction of the jet's motion of

$$P=(1+\cos\beta)\frac{WV}{g},$$

where $\beta$ is the angle between the directions of the jet when entering and leaving the vane. For the surface in motion, $V-u$ is to be substituted for $V$ and the equation becomes

$$P=(1+\cos\beta)\frac{W(V-u)}{g}.$$

The weight of fluid, $W$ pounds per second, is supposed to all act upon the vane.

At the point of exit of the jet from the vane, Fig. 8, lines may be drawn representing $u$ and $V-u$ in magnitude and direction. The diagonal $V_1$ represents in magnitude and direction the absolute velocity of the jet as it leaves the vane.

The impulse of the jet as it enters the vane, in the direction of motion of the vane, is $\frac{WV}{g}$; and as it leaves the vane the impulse is $\frac{WV_1 \cos \Delta}{g}$ in the same direction. Therefore the pressure in the direction of motion of the vane is

$$P=\frac{W}{g}(V-V_1\cos\Delta).$$

But $V_1 \cos \Delta = u-(V-u)\cos\beta$, and therefore

$$P=\frac{W}{g}(V-u)(1+\cos\beta).$$

When $\beta=180°$ there is no pressure exerted upon the vane, and the pressure becomes a maximum when $\beta=0$, for this causes a complete reversal of the direction of motion of the jet.

When the jet strikes the vane as in Fig. 9, at an angle $\alpha$ with the direction of motion of the vane, the stream traverses the surface of the vane with a relative velocity v, found by combining $u$ and $V_1$, and finding their component along the surface of the vane at entrance. The velocity upon leaving

the vane is also v, shown making an angle $\beta$ with the direction of motion of the vane. The absolute velocity of the jet as it leaves the vane is $V_2$.

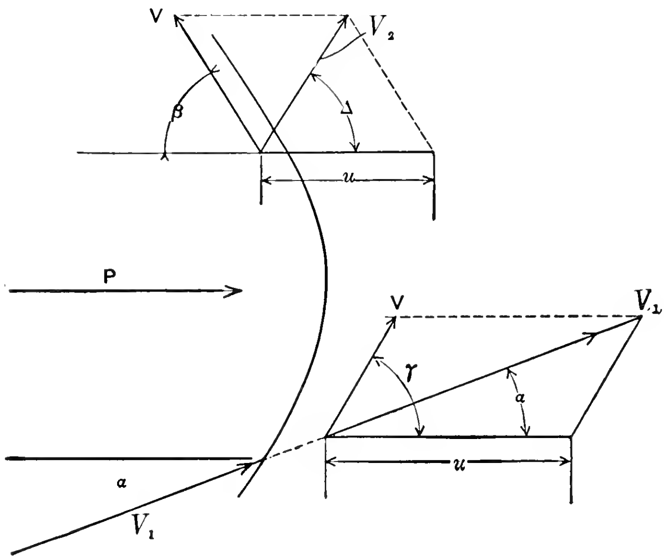



FIG. 9.

The impulse with which the jet strikes the vane is $\frac{WV_1}{g}$ and its component in the direction of motion of the vane is $\frac{WV_1}{g} \cos \alpha$. As the jet leaves the vane the impulse is $\frac{WV_2}{g}$ and its component in the direction of motion of the vane is $\frac{WV_2}{g} \cos \Delta$.

The total impulse in the direction of motion of the vane is

$$P = \frac{W}{g}(V_1 \cos \alpha - V_2 \cos \Delta).$$

*Example 6.*—Let $\alpha = 30°$. $\beta = 40°$.

Let $V_1 = 3000$ ft. per sec. and $u = 1000$ ft. per sec. Then

$$V_2 \cos \Delta = u - \mathrm{v} \cos \beta,$$

and

$$P = \frac{W}{g}(V_1 \cos \alpha - u + \mathrm{v} \cos \beta).$$

The value of v may be found from the lower velocity diagram; thus

$$\mathrm{v} = \sqrt{u^2 + V_1{}^2 - 2uV_1 \cos \alpha}$$

$$= \sqrt{(1000)^2 + (3000)^2 - 6{,}000{,}000 \times .866} = 2192 \text{ ft. per sec.}$$

$$P = (3000 \times .866 - 1000 + 2192 \times .766)\frac{W}{32.2} = 100W, \text{ approx.}$$

If $W = 1$ pound per second, then the impulse produced upon the vane is 100 pounds.

The direction of the line representing the velocity of the steam *relatively* to the vanes or blades of a turbine should be such that the stream or jet enters the blade tangentially to its working face. Otherwise losses due to impact and friction will be greater than necessary.

*Note.*—The difference between the meanings of *impact* and *impulse* should be noted. *Impact* results in loss due to friction between the particles of fluid themselves, or between the fluid and some object upon which it impinges. *Impulse* refers to the dynamic pressure exerted upon some object, as a vane, by a jet possessing kinetic energy. The term *impact-wheel* is therefore a misnomer when applied to turbines used for obtaining useful transformations of energy.

If the jet is to enter the blade tangentially to its surface, the curve of the blade at the edge where the jet enters should be tangent to the line of relative velocity v.

If the angle $\alpha$ is given, $\gamma$, Fig. 9, may be found from the equation

$$\frac{\sin(\gamma - \alpha)}{\sin \gamma} = \frac{u}{V_1},$$

from which

$$\cot \gamma = \cot \alpha - \frac{u}{V_1 \sin \alpha}.$$

Thus the proper value of $\gamma$, the angle of the blade at the entering edge, can be found when $u$, $V_1$, and $\alpha$ are given.

**Work Done by the Fluid Acting against the Vane or Bucket.** Neglecting leakage past the blades of a turbine, all the steam passing through it acts to produce rotation. If the steam enters in the direction of motion of the blades (the latter is not the case in most steam-turbines), leaving at an angle $\beta$ with the direction of motion, the pressure resulting in the direction of motion is

$$P = (1 + \cos \beta)\frac{W}{g}(V_1 - u).$$

The velocity of the blades being $u$, the work done per second is

$$Pu = \left( \left\{ (1 + \cos \beta)\frac{W}{g}(V_1 - u) \right\} \right) u.$$

If $u$ is zero, the work becomes zero, while it becomes a maximum when $u = \frac{V_1}{2}$, or when the linear velocity of the blades is half that of the jet. Making $u = \frac{V_1}{2}$ in the above equation, the work done at the wheel $= (1 + \cos \beta) W \frac{V_1^2}{4g}$.

Dividing by the energy of the jet, $W\frac{V_1^2}{2g}$, the efficiency of the jet is

$$E = \frac{1 + \cos \beta}{2}.$$

Assuming the jet to enter the blades as stated above, the efficiency is seen to depend entirely upon $\beta$, the angle of exit

from the blades. When $\beta=180°$, $E=0$; when $\beta=90°$, $E=.5$; and when $\beta=0°$, $E=1$.

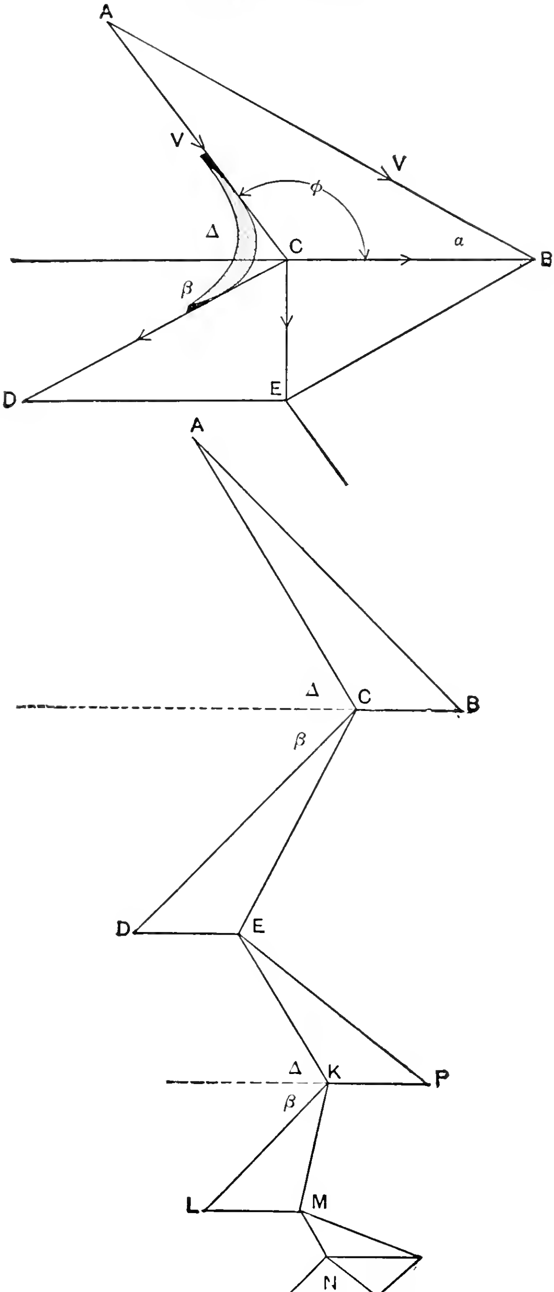
In general, the efficiency of a turbine depends upon the relation between the speed of blade and that of the entering jet of fluid, of whatever kind the latter may be. Assuming that entrance and exit angles are favorable, the highest efficiency may be expected when the speed of blade is from one third to one half the speed of the entering jet. This ratio for highest efficiency, however, depends upon the action of the fluid, whether it works by impulse alone, or by reaction alone, or by both.

Referring to Fig. 10 on page 22, let $AB$ represent in magnitude and direction the absolute velocity, or the velocity relatively to the earth, of the entering steam. Let $CB$ represent the peripheral velocity of the vanes or blades of the turbine. Then $AC$ will represent the velocity of the entering steam relatively to the blades, and $\Delta$ will be the proper blade angle. If the blade curve makes this angle with the direction of motion of the blade, no shock will be experienced when the steam enters the blade. Let the angle at which the steam leaves the blade be $\beta$. Then the absolute velocity of the departing steam is represented by $CE$.

A blade may be sketched in at $C$, Fig. 10, making angles $\Delta$ and $\beta$ with the direction of motion of the blade, and for given values of $\alpha$ and $\beta$, and for a known weight of steam flowing per second, and a known peripheral velocity of blade, the pressure on the blade can be computed as was done in Example 6.

For the compounded turbine the same method may be extended, as shown in Fig. 11. $AB$ and $NP$ represent respectively the initial and final absolute velocities of the steam, and the energy given up by the steam will be proportional to the difference of their squares. Further discussion of this arrangement will be given later.

The preceding discussion illustrates the method by which problems concerning the action of jets upon turbine vanes or buckets may be analyzed. The motion of the vane has been




Figs. 10 and 11.

assumed to be in a straight line, and this assumption will be made in constructing velocity diagrams. The methods to be used are simpler than the preceding, but the work that has been given is useful in showing the general character of the action between the buckets and the working fluid.

**Efficiency of the Impulse-turbine.**—Let steam enter and leave a turbine-bucket (Fig. 12) with relative velocities $v$ and $v_1$ respectively, and let $v = v_1$. Let $\beta = \delta$. The jet enters the turbine-casing at an angle $\alpha$ with the direction of motion of the buckets, and the entering absolute velocity is $V$. The absolute exit velocity is then $V_1$, since the bucket moves with peripheral velocity $u$.

The energy of the entering jet is $\frac{V^2}{2g}$,* and that of the departing jet is $\frac{V_1^2}{2g}$. The work done upon the bucket is therefore

$$W = \frac{V^2 - V_1^2}{2g}.$$

The velocity diagram as shown in Fig. 13 may be reproduced in different form, as shown in Fig. 14. Revolving $v_1$ about the vertical line $AD$ until $v_1$ coincides with $v$, the line representing $V_1$ will take the position $AC$ at the left of the vertical.

Solving the triangle $ABC$ for $V_1^2$ in terms of $V$ and $u$,

$$V_1^2 = V^2 + (2u)^2 - 4Vu \cos \alpha.$$

The efficiency of action of the jet upon the bucket is equal to the energy given up by the jet divided by the total energy of the entering jet; thus,

$$\text{Efficiency} = \frac{V^2 - V_1^2}{2g} \div \frac{V^2}{2g} = \frac{V^2 - V_1^2}{V^2}$$

$$= \frac{V^2 - [V^2 + (2u)^2 - 4Vu \cos \alpha]}{V^2}$$

$$= \frac{4u}{V}\left(\cos \alpha - \frac{u}{V}\right).$$

* Per pound of steam.

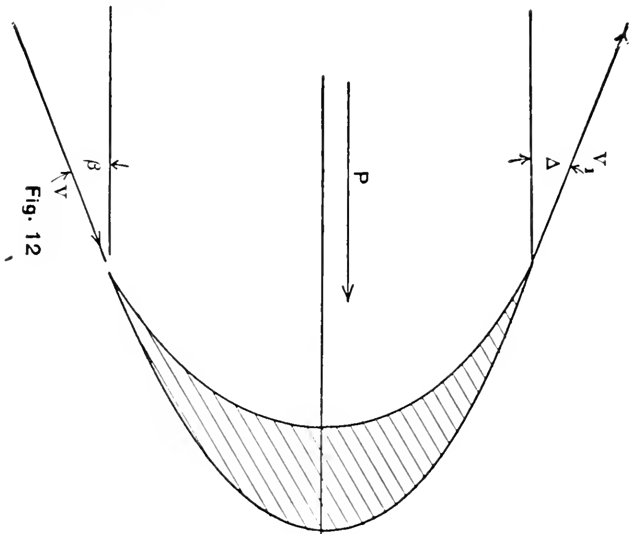


Fig. 12

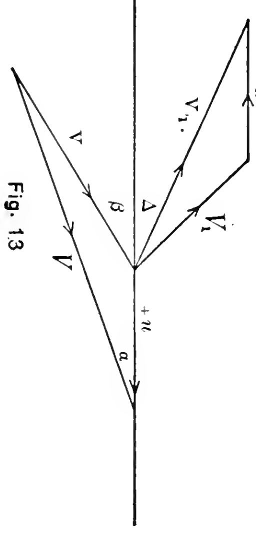


Fig. 13

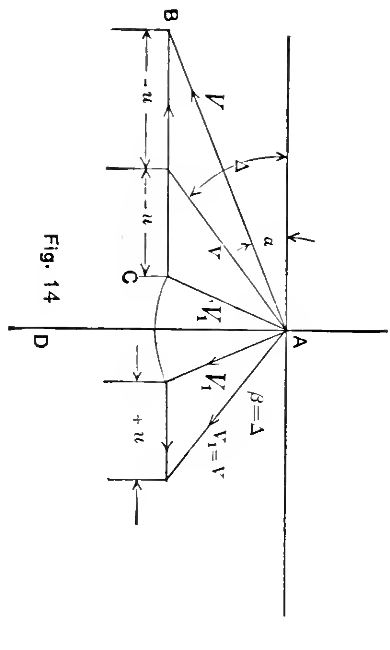
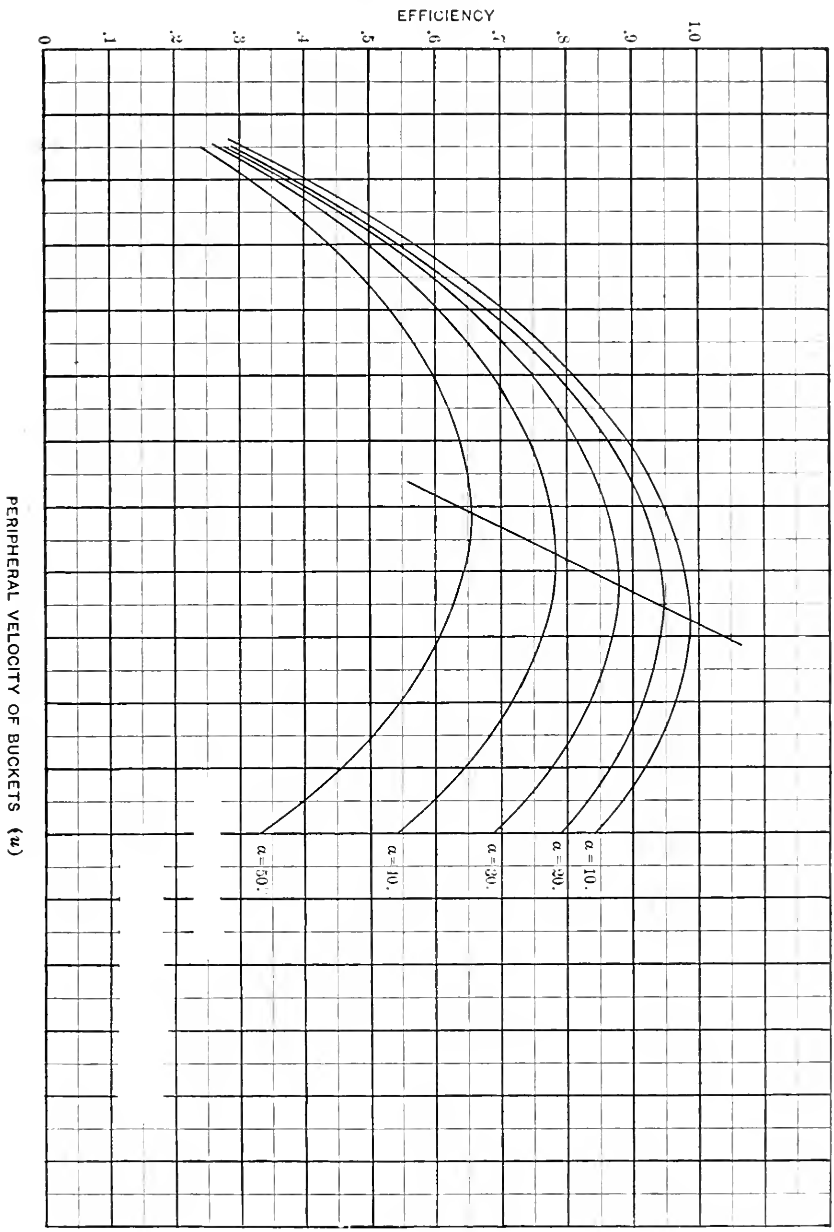


Fig. 14


EFFICIENCY
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
PERIPHERAL VELOCITY OF BUCKETS (u)
α = 50°.
α = 10.
α = 30.
α = 30.
α = 10.

It is evident that the efficiency depends upon the relation between peripheral velocity $u$, entering steam velocity $V$, and the angle $\alpha$ at which the steam leaves the nozzles. If, as is generally the case in the many-stage turbine, the angles of entrance and exit are not equal, the above expression for efficiency requires modification.

The curves on the preceding page show the variation of efficiency for various velocities and angles of entrance of the steam, and the gain accompanying increase of peripheral velocity.

## MEANING OF THE TERMS "IMPULSE" AND "REACTION" AS USED IN THE FOLLOWING CHAPTERS.

Since the forces acting in the two types of turbine are due to two separate although closely related phenomena, it is necessary to give distinctive names to the latter in order to state methods of analsyis.

Reference should be made to pages vii to x in the Introduction, and to page 181 for description, and method employed in solving the problem.

The total dynamic pressure exerted by a stream or jet passing over a blade or bucket surface and experiencing a change in direction of flow, due to the form of the surface, is called *impulse*. Thus the action of fluid upon vanes, as analyzed on pages 10 to 20, results in impulsive pressure entirely. *Reaction*, as used on page 13, is to be understood as meaning that part of the total impulsive pressure upon the surface which is caused by the change into directions of flow having components opposite to the direction of $P$, Fig. 7. $P$ represents the direction in which it is desired to compute the impulsive pressure on the vane. The word *Reaction* need not be used, however, and is not required in the analysis on pages 11 and 12.

*Reaction* is to be understood as the pressure opposite in direction to that of flow, resulting from and accompanying change in the *velocity* of the steam. If the steam falls in pressure during its passage through a row of blades or buckets, its motion is accelerated. This is accompanied by an unbalanced pressure, or reaction, in the direction opposite to that of flow, as described on pages 67–70. In the Parsons turbine impulse and reaction combine to urge onward each moving blade, and in order to analyze the acting forces it is necessary to discriminate between the two methods of producing pressure against the moving elements of the turbine.

# CHAPTER II.

## THERMODYNAMIC PRINCIPLES INVOLVED IN THE FLOW OF STEAM.

When a turbine is operated by steam as a working substance, the steam is so conducted through the machine that it gives up its heat energy in imparting velocity to its own particles. The result is a stream of steam more or less nearly dry, according to the extent to which heat has been changed into mechanical work; and this mass, travelling at high velocity, strikes against the rotating parts of the turbine so as to cause the desired motion.

The preceding chapter deals with the principles of action of a stream or jet as it strikes against and leaves the turbine buckets. The present chapter deals with the methods used for producing the jet or stream of working substance.

The problem before the engineer is, to produce from a given amount of heat energy the greatest possible kinetic energy in a jet of steam issuing in a given direction. This means that a certain weight of steam must attain the highest possible velocity, and that the jet must be conducted in the most efficient manner to the point at which it is to deliver its energy to the buckets or blades of the turbine.

While the design of nozzles and steam-passages is only one among a great many problems before turbine designers, it is of great importance because the efficiency of the nozzle determines the degree of economy with which the heat energy of the steam is changed into mechanical energy. Recent

investigations show that the fundamental thermodynamic equations for the flow of gases must be used with great caution in attempting to predict results of the flow of steam, and that the special conditions under which the steam acts in any given case may be very different from the ideal conditions assumed as the basis for the thermodynamic equations. Further, the equation developed by Zeuner, which has been commonly accepted as applying to steam flow, rests upon the assumption of a constant specific heat of the substance during its expansion, and therefore does not apply in any but a roughly approximate manner to the flow of a varying mixture of steam and water. Coefficients have been worked out by which Zeuner's equation may be modified so as to make it express approximately the results of experiments with different forms of steam orifices and nozzles, but the results have not, so far, led to methods of predicting what may be expected to occur in a given proposed case.

Steam is an elastic fluid, and it has the power of expanding indefinitely as the pressure in the containing space is further and further diminished. This power of expansion is possessed by virtue of the intrinsic energy of the steam, or the energy due to the heat contents of the steam. Work has been done upon the steam in supplying it with heat energy, and the steam is capable of increasing its volume and giving up energy to other bodies of matter as it moves them out of the way, and thus it does what is called *external work*. Also, the steam in expanding experiences changes in its own molecular activity; its temperature and pressure are lowered as it gives up its heat during expansion, and these changes in the internal condition of the steam result in what is called *internal work*. The work done in displacing the surroundings as the steam increases its volume is called external work.

A coiled spring presents similar conditions. When it has been compressed or extended by work done upon it, the spring is capable of changing its length and of exerting force upon other bodies while doing so. The change in the condi-

tion of the parts of the spring itself is called internal work, and the energy it gives up to other bodies is called external work. During a boiler explosion steam does external work in rupturing and displacing the boiler parts and in displacing and vibrating the atmosphere. The steam, finding it possible to fall in pressure and temperature, experiences a change in its internal condition, and this change results from what is called internal work. In this case the internal work is negative, since it is accompanied by a decrease of the internal energy of the steam.

Imagine a gas-tight vessel, containing air or gas at a certain pressure. Let heat be lost by radiation from the walls. The temperature and pressure of the gas will fall, and, in general, internal work will be done in changing the internal energy of the gas. The volume remains constant, and therefore no external work is done.

If the walls, on the other hand, do not transmit heat, and if, instead of the gas being kept at constant volume, an opening is made in the vessel, a flow of gas will occur through the opening and external work will be done upon the outside medium, supposing the pressure in the latter to be lower than that of the gas in the vessel. If, however, the pressure in the vessel be lower than that outside, the outside medium will rush in and do work upon the gas, raising its temperature and pressure.

In the first case, the gas rushes out of the vessel, displacing some of the external atmosphere, thus doing external work,—and it also changes its own temperature and pressure, thus doing internal work. In the second case, the external atmosphere possesses the greater energy and it does external work upon the gas in the vessel, by compressing it into smaller volume; and it does internal work upon it by increasing its temperature and pressure. In both cases heat is expended, and both external and internal work are done. Only in the case of the gas-tight vessel is the work all internal work.

Both internal and external work are done at the expense

of the intrinsic energy of any fluid, whether gas or air or steam, and in general the following equation may be written:

$$\text{Heat expended} = \text{Internal work} + \text{External work}.$$

A given weight of gas at given pressure and temperature occupies a certain known volume, and contains a known amount of heat energy. If the gas be caused to expand at constant temperature, the product of pressure and volume remains constant, or its condition may be found at any point of its expansion from the equation

$$pv = p_1v_1 = p_2v_2, \text{ etc.}$$

In order to obtain such expansion, however, heat must be added to the gas continuously, during its expansion, in just sufficient quantities to restore to the gas the heat equivalent of the work done. The gas gives up, continuously, its internal energy, to overcome whatever external resistance may be opposed to its expansion. Since the gas receives compensation for all energy expended, it possesses the same internal energy at the end of expansion that it did before it commenced to expand. Such a process is known as *isothermal expansion*, and the equation of the isothermal expansion line may be found by making temperature constant in the fundamental equation for gases,

$$\frac{pv}{T} = \frac{p_1v_1}{T_1}, \quad \ldots \ldots \ldots \quad (7)$$

$T$ being the absolute temperature at which expansion occurs.

If expansion takes place from $p_1v_1$ to $p_2v_2$, Fig. 15, the external work is represented by the shaded area beneath the curve $pv = p_1v_1$, and equals

$$W = p_1v_1 \int_{v_1}^{v_2} \frac{dv}{v} = p_1v_1 \log_e \frac{v_2}{v_1}.$$

It is shown in thermodynamics that if a gas expands adiabatically,—that is, without receiving or giving out heat, as heat,—the equation to the expansion curve may be written

$$pv^n = p_1 v_1{}^n = p_2 v_2{}^n, \text{ etc.,}$$

where $n$ is the ratio of the specific heats of the gas at constant pressure and at constant volume respectively.

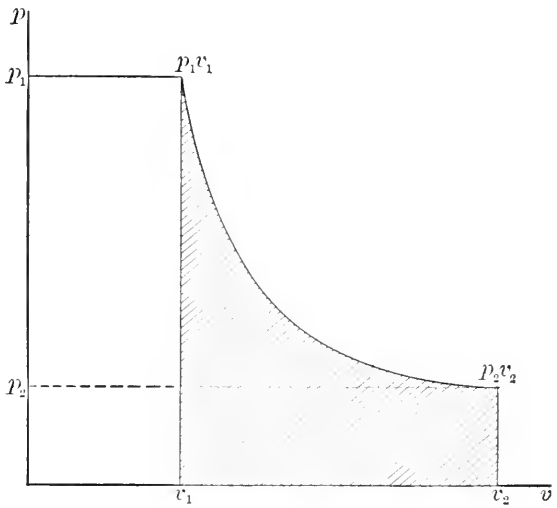



FIG. 15.

Let a quantity of gas be at the state $p_1v_1$ (Fig. 16) and let it expand to $p_2v_2$ adiabatically. The external work is

$$W = \int_{v_1}^{v_2} p dv = p_1 v_1{}^n \int_{v_1}^{v_2} \frac{dv}{v^n}$$

$$= \left(\frac{1}{v_2{}^{n-1}} - \frac{1}{v_1{}^{n-1}}\right)\left(\frac{p_1 v_1{}^n}{n-1}\right)$$

$$= \frac{p_1 v_1}{n-1}\left\{1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right\}.$$

As no heat is supplied to the gas during expansion, the external work possible is limited in amount according to the

intrinsic energy of the gas at $p_1v_1$. The capacity of the gas to do work is measured by the area beneath the curve, extended indefinitely to the right, and the axis of volume. When the volume becomes indefinitely great the gas has done all the external work it is capable of doing. Since $v_2$ has become

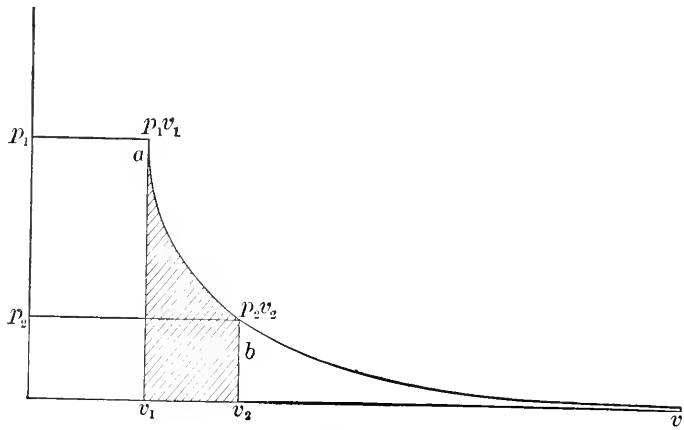



FIG. 16.

indefinitely great, $\frac{v_1}{v_2} \doteq 0$, and the expression for the work done becomes simply

$$W = \frac{p_1v_1}{n-1}.$$

This measures the total intrinsic energy of the gas, or working substance.

The intrinsic energy of the gas at $a$ is

$$E_1 = \frac{p_1v_1}{n-1},$$

and at $b$ it is

$$E_2 = \frac{p_2v_2}{n-1}.$$

When a body receives heat, and does not change its state during that reception of heat, its temperature rises, and the body either expands in volume, or its pressure increases. Thus, according to the assumption that rise of temperature means increased vibratory activity of the particles composing the body, the internal kinetic energy is increased. The internal condition of the body is also changed to the extent of increasing the distances between the particles of the body, as the latter expands.

Besides the changes of internal energy, the expansion of the body causes displacement of any substance surrounding it, or opposing its expansion. This is called external work. Due to the increase in the internal or intrinsic energy of the body by the addition of heat, external work is done upon the surroundings of the body by the action of the heat in causing enlargement of the space occupied.

Further, if the substance be a fluid such as gas or steam, held within a vessel and containing a given amount of heat energy, the substance will flow from a properly arranged orifice in the containing vessel, if the orifice opens into a medium of lower pressure than that in the vessel. Thus the energy of the substance will be utilized in a third manner, that of giving velocity to the particles composing the substance and thus increasing its kinetic energy.

Let the vessel *a* be fitted with an orifice at *b*, with well-rounded entrance so that no losses occur due to irregularity of flow at entrance to the orifice. Further, let the orifice present no frictional resistances to the flow of the substance, now supposed to be a gas. Let the intrinsic energy of the gas be called $E_1$ and $E_2$, when inside the vessel and the nozzle respectively. External work $p_1v_1$ is done upon each pound of gas leaving the vessel, and each pound does external work $p_2v_2$ as it expands in the nozzle. The kinetic energy due to the velocities in the vessel and the nozzle respectively are $\frac{V_1^2}{2g}$ and $\frac{V_2^2}{2g}$ per pound of gas.

Now, if the flow of the substance is adiabatic, the total

energy in the gas remains the same at all times during the flow, and may be expressed by the following fundamental equation for the flow of elastic fluids:

$$E_1 + p_1 v_1 + \frac{V_1^2}{2g} = E_2 + p_2 v_2 + \frac{V_2^2}{2g},$$

$v_1$ and $v_2$ representing volumes per pound of the substance at pressures $p_1$ and $p_2$ respectively; while $V_1$ and $V_2$ repre-

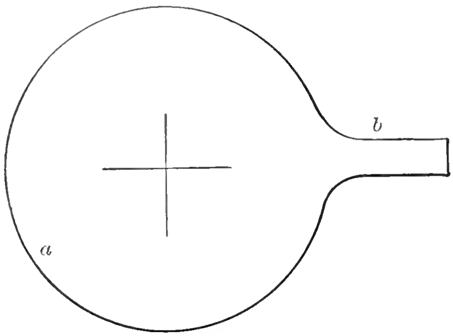



FIG. 17.

sent velocities. The velocity $V_1$ in the vessel is usually negligibly small compared with $V_2$, and suppressing $\frac{V_1^2}{2g}$, the equation becomes

$$\frac{V^2}{2g} = E_1 - E_2 + p_1 v_1 - p_2 v_2. \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Since the right-hand member of the equation represents the sum of the change in internal energy and the external work done upon and by the substance during its expansion from $p_1 v_1$ to $p_2 v_2$, and since the changes have been due solely to the work done by the heat energy in the steam, it follows that the resulting kinetic energy, $\frac{V^2}{2g}$, per pound of the issuing stream, is numerically equal to the amount of heat

each pound of the substance has given up during its expansion from $p_1v_1$ to $p_2v_2$.

If the total heat of the substance at $p_1v_1$ be called $H_1$, and that at $p_2v_2$ be called $H_2$, then for each pound of the substance the energy of the jet flowing from the nozzle is

$$\frac{V^2}{2g}=(H_1-H_2)\times 778.\ \text{foot-pounds.} \quad . \quad . \quad . \quad (9)$$

From this equation may be calculated the velocity that would result in an ideal case from a given fall in heat contents of a known quantity of gas or steam, if the flow were confined to a given direction.

*Example.*—Steam flows through a nozzle, and in doing so falls in pressure to such an extent as to make a difference of 225 thermal units per pound between the initial and final heat contents. Calculate the resulting velocity, assuming that there are no losses of energy in the nozzle.

One thermal unit = 778. foot-pounds of energy.

$$H_1-H_2=225.\ \text{B.T.U.}$$

$$V=\sqrt{778.\times 225.\times 64.4}=3360.\ \text{ft. per second.}$$

**The following development of Zeuner's equation** is given because, while it does not apply exactly to the flow of steam, it is of considerable interest in all thermodynamic work, and it does apply directly to the flow of a fluid the value of whose ratio of specific heats, at constant pressure and constant volume respectively, does not change during the flow. It is of particular interest since it indicates that, after a certain diminution of the lower pressure in the case of the flow of a substance from a higher pressure to varying lower pressures, the rate at which the substance flows does not increase. The rate increases until the ratio of final to initial pressure reaches a certain value, after which no further increase accompanies a further lowering of the final pressure. The equation is that of a

curve which reaches a maximum, after which it decreases to zero. (See curves No. 5, on pages 96, 97, 98).

Equation 8 may be written

$$\frac{V^2}{2g}=\frac{p_1v_1}{n-1}-\frac{p_2v_2}{n-1}+p_1v_1-p_2v_2=\frac{n}{n-1}(p_1v_1-p_2v_2),$$

in which $n$ represents the ratio of the specific heats of the substance at constant pressure and constant volume respectively.

Remembering that $p_1v_1{}^n=p_2v_2{}^n$,

$$p_2v_2=p_1v_1\left(\frac{v_1}{v_2}\right)^{n-1}=p_1v_1\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}.$$

from which

$$\frac{V^2}{2g}=p_1v_1\left(\frac{n}{n-1}\right)\left\{1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right\},$$

and

$$V=\sqrt{2gp_1v_1\left(\frac{n}{n-1}\right)\left\{1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right\}}.$$

If the area of the orifice is $a$, the volume emitted per second $=aV$ and if $v_2$ is the specific volume at pressure $p_2$, the weight discharged per second is

$$W=\frac{aV}{v_2}.$$

But

$$v_2=v_1\left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}$$

Therefore

$$\text{Weight per second}=W=a\sqrt{\left(\frac{2gp_1}{v_1}\right)\frac{n}{n-1}\left\{\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}}-\left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right\}}.\quad(10)$$

Let $\frac{p_2}{p_1}=r$. Then the weight $W$ becomes a maximum when $(r)^{\frac{2}{n}}-(r)^{\frac{1+n}{n}}$ becomes a maximum.

Differentiating with respect to $r$, and equating to zero,

$$\frac{2}{n}(r)^{\frac{2}{n}-1}-\left(1+\frac{1}{n}\right)r^{\frac{1}{n}}=0.$$

Dividing by $(r)^{\frac{1}{n}}$,

$$r=\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}.$$

The value of the ratio $r\left(=\frac{p_2}{p_1}\right)$ for maximum flow of air under adiabatic conditions is 0.528, the value of $n$ being 1.41. For dry saturated steam the ratio of specific heats is ordinarily taken as 1.135 which gives a maximum flow, by weight, when $\frac{p_2}{p_1}=0.577$.

The above equation (No. 10) is plotted on Plates IV, V, and VI, and the curve indicates that if the pressure in the receiving vessel should be reduced to zero, the weight of fluid discharged by the orifice or nozzle per unit of time would be zero. It was stated on page 35 that the reasoning upon which the equation was developed applied to substances within the limits of pressure and temperature pertaining to a given physical state, in which the ratio of specific heats, $n$, remains constant. The reasoning is correct, and experimenters have met with some, though not complete, success in attempting to verify the conclusions regarding adiabatic expansion of gases.* It has been demonstrated experimentally that air, and that

* See paper by Wm. Froude, "Engineering," London, 1872; also paper by Professor Fliegner, Zeitschrift des Vereines d. Ingenieure, 1896.

gases in general, in flowing from higher to lower pressures through orifices, increase their weight of flow per unit of time as the back pressure $p_2$ is reduced, but that after reduction of $p_2$ to about 0.52 $p_1$ no further increase in rate of flow can be brought about by further reduction of $p_2$*

The experiments of Professor Gutermuth, plotted upon Plates IV, V, and VI, show that the weight of steam discharged per second does reach a maximum, as the equation indicates that a perfect gas should do, but that the flow of steam, instead of decreasing in rate after the maximum has been reached, remains constant no matter how much the back pressure be further reduced.

*If the lower pressure, $p_2$, be kept constant*, and the initial pressure be increased, the rate of flow, by weight, will increase in direct proportion to the increase in initial pressure. Experimental evidence as to this and as to the statements made in the preceding discussion will be given during the development of the subject of the flow of steam.

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* See bottom of page 62.

## CHAPTER III.

### GRAPHICAL REPRESENTATION OF WORK DONE IN HEAT TRANSFORMATIONS.

THE pressure-volume diagram, of which the ordinary steam-engine indicator card is an example, and the heat diagram, or, as it is generally called, the temperature-entropy diagram, are two means by which the effect of transforming heat into mechanical work is represented. The present chapter will discuss the heat diagram, which serves a purpose distinct from that of the work, or pressure-volume diagram. Either method of representation taken alone is incomplete without the other, while the two together completely satisfy the requirements in analyzing graphically a thermodynamic problem from an engineering standpoint.

In Fig. 18 let ordinates represent absolute temperature. It is required to construct a diagram whose area shall represent heat quantities in thermal units, and absolute temperature is required to be used as one dimension of the heat represented by the diagram. This is done because temperature is the intensity factor of a heat quantity, and absolute temperature is used because the fundamental laws of thermodynamics are, as they are now understood, based upon the scale of absolute temperature. The adoption of this scale in the heat diagram thus relates computations made from the diagram to those made by the laws of heat as ordinarily expressed. It is required to find another function which taken as an abscissa in connection with absolute temperature as an ordinate will give

a diagram whose area represents heat-units, as described above. It is well in approaching the heat diagram for the first time to start without any thought of entropy, unless one has a very

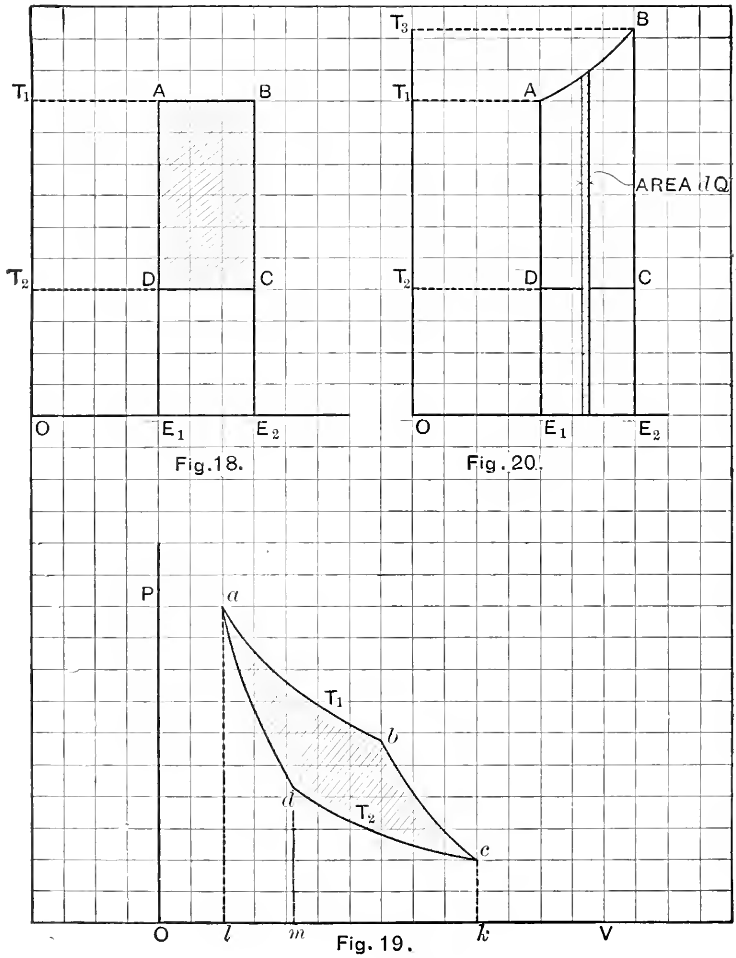



Fig. 18.

Fig. 20.

Fig. 19.

clear notion of the meaning of that word, and to simply determine for one's self the character of the abscissa of the heat diagram. This will later be found to be the same as the function to which the name entropy was given by early investigators of the science of heat.

Let the quantity which is to be represented by abscissa always increase when heat, as heat, is added to a substance, and decrease when heat, as heat, is taken away. A vertical line then represents a set of conditions in which the temperature changes, but during the change there is no heat, as heat, given to or taken away from the substance. This is what is called an *adiabatic process*, which means that no heat, as heat, has been given to or taken away from the working substance during the process. In other words, the vertical line is what is called an *adiabatic*.

While the word "adiabatic" means that no heat communication takes place between the working substance and other bodies during the process in question, there is always work done when a substance expands against a resistance, and this work is done at the expense of the heat energy possessed by the body. Therefore during adiabatic expansion heat does leave the substance as work done, but not in the form of heat. The adiabatic curve in the pressure-volume diagram, and the vertical or adiabatic line in the heat diagram, represent a change during which work is done, and therefore the intrinsic energy of the working substance is diminished; but during the process no heat has been given to or taken from the working substance, excepting as heat has been transformed into mechanical energy. A horizontal line represents a process during which heat is added to or abstracted from a substance at a constant temperature; that is, there is no temperature change during the process. A horizontal line then represents in the diagram what is called an isothermal change, or a change at constant temperature, and the function which is to be found and used as abscissæ in the diagram is the scale by which the relation between different adiabatic changes is expressed. Thus in Fig. 18, $AD$ represents an adiabatic change in which a substance whose temperature was originally that represented at the height $A$ has fallen in temperature to $D$ without having received or given up any heat as heat. The line $BC$ represents a similar adiabatic drop in

temperature. The horizontal line $AB$ is a line of constant temperature, and the distance $AB$ or $E_1E_2$ represents the change of abscissa corresponding to a change in heat contents measured by the area $ABE_2E_1$. $AB$ is what is called an isothermal line, and a quantity of heat represented by the area $ABE_2E_1$, under the line $AB$ and extending to the line of zero absolute temperature, has been added to the substance, thereby moving the point representing the state or condition of the substance, from $A$ to $B$. The state of the substance, represented by the point $A$, shows that its temperature is $T_1$. The method by which this temperature was attained is not shown, and it is not necessary that it be known in order that the effect of further operations may be represented. If heat is added to the substance isothermally, the state point will move from $A$ to $B$, and the distance $AB$ will be such that the heat that has been added equals the area $ABE_2E_1$.

To make the above clear, suppose in Fig. 19 the ordinates and abscissæ represent pressure and volume respectively. Then the familiar Carnot cycle will be represented by two isothermals $ab$ and $cd$ intercepted by two adiabatics $bc$ and $da$. The cycle is represented in Fig. 18 by the figure $ABCD$. The mechanical equivalent of the heat involved in the cycle Fig. 19 is represented by the area $abckl$, and in the heat diagram Fig. 18 the heat involved in the process is represented by $ABE_2E_1$. The mechanical equivalent of the heat rejected at the lower temperature $T_2$ is represented in Fig. 19 by the area $cdmk$, and in Fig. 18 the heat is represented by the area $CDE_1E_2$. The shaded area in each of the figures represents the net work accomplished during the cycle. In the heat diagram the area $ABCD$ represents heat-units utilized during the cycle, and in the pressure-volume diagram, Fig. 19, the area $abcd$ represents the work realized in foot-pounds. The efficiency of the cycle represented in Fig. 19 is

$$\frac{T_1-T_2}{T_1},$$

and it is easy to see that the cycle represented in Fig. 18 has the same efficiency; that is, the shaded area $ABCD$ divided by the area $ABE_2E_1$ is the efficiency of the cycle, and this obviously equals

$$\frac{T_1 - T_2}{T_1}.$$

If the total heat beneath the line $AB$, Fig. 18, that is, the heat $ABE_2E_1$, equals $Q$, then the heat transformed into useful work during the cycle equals

$$\frac{Q}{T_1} \times (T_1 - T_2).$$

The quantity $\frac{Q}{T_1}$ is obviously a measure of the distance $E_1E_2$, or it is what is commonly called the increase of entropy occurring between the initial and final states $A$ and $B$ respectively. For an isothermal change, then, the change in entropy is equal to $\frac{Q}{T}$, where $T$ represents the absolute temperature at which the heat $Q$ is received.

Absolute quantities of entropy are not measured, but only the differences of entropy between two states of a substance, as the total value of the entropy above absolute zero is not known, and is not necessary for engineering purposes.

Suppose that the state of a substance is represented (Fig. 20) by the point $A$, and heat be added to the substance, raising its temperature. The substance may be considered to be any solid which is heated without experiencing a change in its state, as from solid to liquid, liquid to gaseous, etc., or it may be a gas supposed to not change its state during the heat change under consideration. In Fig. 18 heat was added isothermally, as when a substance like water is evaporated, along the line $AB$; but if at the point $A$ (Fig. 18) the substance had been water below its boiling temperature, then if heat had been added to

it, a rise in temperature would have occurred along some such curve as $AB$ (Fig. 20). Now, let the heat diagram that is to be constructed be such that the area underneath any line, as the line $AB$, down to absolute zero of temperature, represent the total heat involved in the process; then the heat added to move the state point from $A$ to $B$ is that represented by the area $ABE_2E_1$, Fig. 20. If one pound of the substance is supposed to be involved in the process, having a specific heat of $S$, then the heat that caused the rise in temperature from $T_1$ to $T_3$ is represented by the area $ABE_2E_1$ and is equal to $S(T_3-T_1)$. This follows from the definition of specific heat. Let the quantity of heat be called $Q$, as was done in the case of the isothermal addition of heat along $AB$ in the discussion of Fig. 18. It is desired to do for the diagram in Fig. 20 just what was done for that in Fig. 18, that is, to find the increase in the value of the abscissa due to the addition of the heat $Q$. In the case of the rectangular diagram of Fig. 18 it was a simple matter to divide the area of the rectangle by one dimension, or the increase of the abscissa $E_1E_2$. This was found to be $\frac{Q}{T_1}$, and this quantity multiplied by the temperature range $(T_1-T_2)$ gave the total heat utilized during the cycle. In Fig. 20 the cycle begins with an addition of heat to a body having absolute temperature $T_1$. The result is a rise of the temperature of the body to $T_3$ and a change of position on the diagram of the state point to $B$. The quantity of heat $Q$ causing this rise is represented by the area between $AB$ and the line of zero temperature, that is, by the area $ABE_2E_1$. The next step in the cycle is an adiabatic expansion of the body from $T_3$ to $T_2$, and this expansion is represented by the vertical line $BC$. Just as in the Carnot cycle of Fig. 18, heat is rejected or exhausted along the isothermal $CD$, and the body is brought to its original condition at $A$ by an adiabatic compression along the vertical line $DA$. The only difference between the two cycles is that heat was added isothermally in that of Fig. 18, and with a rising temperature in Fig. 20.

Returning to the equation last written, the quantity of heat added is

$$Q = S(T_3 - T_1).$$

This, however, gives no clue to the amount by which the abscissa of the diagram has been increased, and it is this quantity which is required in order to make it possible to trace out the path by which the state point moved from $A$ to $B$ during the addition of the heat $Q$.

The area $ABE_2E_1$ may be divided into very small areas, similar to the area $dQ$ in Fig. 20, and if the width of each of these is given the indefinitely small value $dE$, then the vertical height, or the absolute temperature at which the heat represented by $dQ$ is added, may be considered as constant during the addition of the heat $dQ$. An equation may then be written thus:

$$dQ = TdE,$$

where $T$ represents the absolute temperature at which $dQ$ is added to the substance.

Similarly the equation

$$Q = S(T_3 - T_1)$$

may be made to express the heat represented by the area $dQ$ by making use of the fact that during the addition of $dQ$ the rise of temperature is only an infinitesimal amount $dT$ instead of $(T_3 - T_1)$. The expression thus becomes

$$dQ = SdT.$$

The two expressions for $dQ$ may now be equated thus:

$$dQ = TdE = SdT$$

or

$$dE = S\frac{dT}{T}.$$

The distance $E_1E_2$ is equal to the sum of all the small distances like $dE$, and therefore the distance $E_1E_2$ or the total increase of the abscissa of the state point during the change of the temperature of the substance from $T_1$ to $T_3$ is equal to the summation of all the quantities $S\frac{dT}{T}$ between the limits of temperature $T_1$ and $T_3$. Expressing this in mathematical form

$$E=S\int_{T_1}^{T_3}\frac{dT}{T}=S\log_e\frac{T_3}{T_1}.$$

Stating briefly the substance of the preceding discussion:

I. The ordinate of the point representing the state of the working substance as to temperature and heat changes increases and decreases as the absolute temperature of the substance rises and falls.

II. The abscissa of the state point increases and decreases during addition and abstraction of heat respectively, and the amount by which it changes is expressed in the two following ways:

(*a*) The increase or decrease is

$$E=\frac{Q}{T_1},$$

when heat is added or abstracted at a constant temperature $T_1$, as in the boiling of water and the condensation of steam.

(*b*) The increase or decrease is

$$E=S\log_e\frac{T_3}{T_1},$$

when heat is added or abstracted and thereby raises or lowers the temperature of the substance from $T_1$ to $T_3$, as in the heating or cooling of a gas between such limits of temperature that the physical state of the gas does not change in the process.

In the above, $S$ is the mean specific heat of the substance between the temperatures $T_1$ and $T_3$.

Let a pound of water be at 493 degrees F. absolute temperature, corresponding to about 32 degrees on the ordinary Fahrenheit scale. The water is then at the temperature at which ice melts. As a matter of convenience the tables giving the properties of water and steam have been commenced at this temperature. Since the total value of the entropy of the substance is not used in computation, but only the increases or diminutions of entropy due to additions and abstractions of heat respectively, the line representing zero entropy may be located in any convenient position. Steam-engine problems are ordinarily concerned with the properties of water and steam above the melting-point, and therefore the line of zero entropy may be conveniently placed so as to disregard the heat that exists in the water before it reaches the temperature

*Reason for the use of the term Entropy.*

The expressions $\frac{Q}{T}$ and $\int\frac{dQ}{T}$ have been used since the researches of Clausius and Rankine, and are of fundamental importance in analyzing heat problems.

The name Entropy was applied by Professor Clausius to the general expression $\int\frac{dQ}{T}$, and Professor Rankine called it "The Thermodynamic Function." Rankine used the Greek letter $\phi$ to represent the function, and various writers using the Greek letter $\theta$ to represent absolute temperature have called the heat diagram "The Theta-phi-diagram." The name generally given to it, however, is "The Temperature-entropy Diagram." A discussion of reversible and irreversible processes is involved in satisfactorily explaining the meaning and application of the term "Entropy," and for such discussion recourse may be had to the works of Clausius, Zeuner, Rankine, and other writers upon thermodynamics. The following articles discuss the recent literature of the subject:

"On Clausius' Theorem for Irreversible Cycles, and the Increase of Entropy," by W. McF. Orr, Philosophical Magazine, Vol. 8, 1904, page 509.

"On Certain Difficulties which are Encountered in the Study of Thermodynamics," by Dr. Edward Buckingham, Phil. Mag. Vol. 9, 1905.

of melting ice at the mean barometric pressure. The diagram on Plate I must be imagined to extend below the line of 490 degrees absolute down to absolute zero. The total area beneath any line representing a continuous change in the condition of the substance, and down to absolute zero of temperature, represents the British Thermal Units involved in the change.

The curve $XAB$ represents the addition of heat to water, thus raising its temperature from that of melting ice to higher temperatures. The increase in entropy from 493 to 750 degrees is approximately

$$E_B = S \log_e \frac{T_1}{T_2} = 1 \times 0.42.$$

The entropy of the point $B$ is seen to correspond with this value.

The specific heat of water, $S$, is not constant, and on a rigid computation for change of entropy over a range of temperature it is necessary to take the mean specific heat for the temperature range in question. Within the limits just used the mean value for $S$ is 1.006, or very nearly unity. In steam-engine problems in general the value of unity may be used without any greater error than is always involved in reading results during engine tests The total heat above that at freezing-point in the pound of water at $B$ is

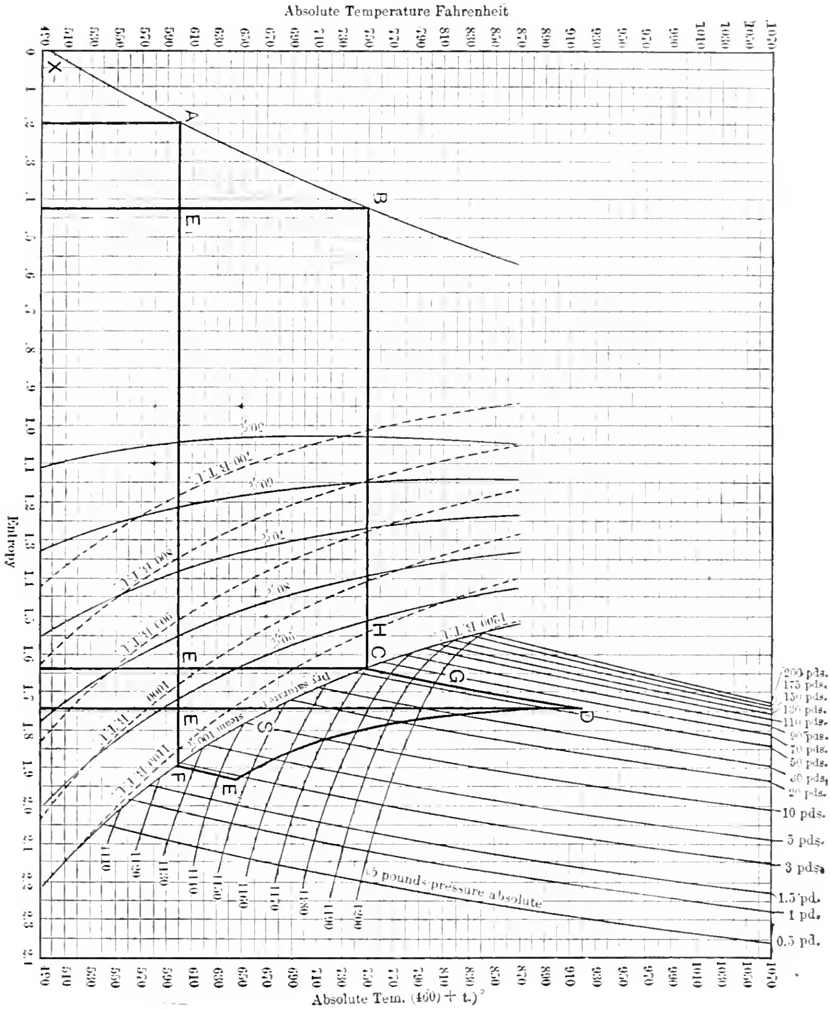
$$H_B = S(T_1 - T_2) = 1.006(750 - 493) = 258.6 \text{ B.T.U.}$$

By looking in the steam-tables for the heat of the liquid above 32 degrees corresponding to 750 degrees this value will be found.

The curve $AB$ represents the heating and cooling of water, and its equation is

$$\text{Change of entropy} = S \log_e \frac{T_1}{T_2}.$$

PLATE I.


Absolute Temperature Fahrenheit
Entropy
Absolute Tem. (460) + t.)°
X
A
B
E
H
C
G
D
E
E
F
E
S
200 pds.
175 pds.
150 pds.
130 pds.
110 pds.
90 pds.
70 pds.
50 pds.
30 pds.
20 pds.
10 pds.
5 pds.
3 pds.
1.5 pd.
1 pd.
0.5 pd.
.5 pounds-pressure absolute
Dry saturated
steam 100%
900 B.T.U.
1000 B.T.U.
1100 B.T.U.
1200 B.T.U.

The line $BC$ is an isothermal, or line of constant temperature, and represents the addition of the heat of vaporization to water of the temperature represented by the height of the line. Water at $B$ is just ready to become steam, and a slight addition of heat generates a correspondingly small quantity of steam.

By experiment it has been found that if to the pound of water at 750 degrees absolute temperature there be added about 911 heat-units the water will be completely evaporated into dry steam. The total heat above 32 degrees would then be

$$258.6+911=1169.6.$$

By consulting the steam-tables this will be found to be the value given for the total heat above 32 degrees of the ordinary scale, or above 493 degrees absolute.

If only half of 911 heat-units had been added to the water at $B$ only half a pound of steam would have been formed, or the "quality" of the steam would have been 50%. It will be found by measurement that the curve on the diagram marked 50% divides each horizontal distance such as $BC$ into two equal parts. Similarly, the curve marked 90% divides the distance into parts which are to each other as 9 is to 1. This means that if the addition of heat at a given temperature should be stopped at the intersection of this curve with the horizontal line representing the given temperature, 90% of the heat necessary to evaporate a pound of water into dry steam would have been added, or there would be produced 0.9 pound of steam. The remaining 0.1 would remain as water, either in the boiler or suspended in the steam.

The curve $CF$ is called the "Saturation Curve," and is drawn through the extremities of the horizontal lines representing the increase of entropy accompanying the addition of sufficient heat at different temperatures to completely vaporize a pound of water at these temperatures.

The area beneath the line $BC$, down to absolute zero of temperature, represents the heat of vaporization or the "latent heat" of a pound of steam at the temperature 750 degrees absolute. The increase of entropy between $B$ and $C$ is found by dividing the heat of vaporization by the absolute temperature at which it is added, or

$$E_{BC}=\frac{911}{750}=1.215.$$

This may be verified by subtracting $E_B$ from $E_c$ on the diagram.

The curves marked 1100 B.T.U., 1000 B.T.U., etc., cut the horizontal lines in such points that if the addition of heat should be stopped at these intersections the pound of steam and water would contain the amount of heat indicated by the figures on the curve. Thus, if heat were added along $BC$ till the entropy increased to that at $H$, the pound of steam and water would contain 1100 B.T.U. above the temperature of melting ice. The fraction $\frac{BH}{BC}$ of the total heat of vaporization present is, approximately,

$$H_v=\frac{1.12}{1.21}\times 911=842+$$

| | | |
|---|---|---|
| Heat of liquid | $H_B=$ | 258+ |
| Total heat above freezing | | 1100 B.T.U. |

If heat be added to the steam after it has become dry and saturated, as at $C$, the result is the production of what is called "*superheated steam.*" As heat is added to it, the temperature rises; that is, the "degrees of superheat" increase. Superheated steam behaves much as does a gas. The curve $CD$ has the same equation as the curve $AB$, with the exception that the value of the specific heat is different, and the increase of

entropy accompanying an increase of temperature from $T_c$ to $T_D$ is

$$E=S\log_e\frac{T_D}{T_c},$$

where $S$ is the mean specific heat of superheated steam for the range in question.

Taking 0.57 as the specific heat, the increase of entropy during addition of heat from $C$ to $D$ is

$$E_{CD}=0.57\log_e\frac{920}{750}=0.57\times0.199=0.113.$$

This will be found to correspond approximately with the value given on the diagram.

The heat involved in raising the temperature of steam from the saturation temperature at $C$ of 750 degs. to 920 degs. is

$$H_s=0.57(920-750)=0.57\times170=97 \text{ B.T.U., approximately.}$$

The total heat in the superheated steam, then, above 32 degs. F. is

$$258+911+97=1266 \text{ B.T.U.}$$

It will be evident, upon finding the area beneath the broken line $XBCD$ down to absolute zero, or 490 degs. below the base line of the diagram, that this area represents the number of thermal units stated. The dimensions in which the area is measured are the same as those representing *degrees temperature*, and *entropy units*. Thus, the heat under the line $BC$ is represented by an area 1.215 units in width horizontally and 750 units vertically, giving 911 thermal units as the heat so represented.

Curves of constant pressure such as $CD$ are plotted by

the methods of the last example, which give the increase in entropy accompanying any rise in temperature, as from $C$ to $D$. The point $C$ represents the state of a pound of dry steam at normal temperature corresponding to its pressure. The steam contains in that condition a certain amount of heat which is different from the amount contained in a similar amount of dry steam at any other pressure. The line $CD$ represents the addition of heat to the normal amount of heat at $C$. The value of $S$ to be used in the equation for the line $CD$ is the specific heat * of superheated steam at constant pressure; that is, it is the number of thermal units required to raise a pound of steam of pressure corresponding to a certain temperature, by one degree Fahrenheit. If the specific heat is constant for all pressures and temperatures then one value is to be used in all cases. If it changes when the pressure changes then a different value must be used for each pressure. If it changes as the temperature changes, then for a given temperature range a mean value must be found which, when multiplied by the temperature range, will give the quantity of heat required to cause the rise of temperature involved.

In any case, since the superheat indicated by the area beneath $CD$ is the heat necessary to raise dry steam of the temperature and pressure indicated at $C$, and does not apply to the superheat for any other pressure, the line $CD$ is properly called a "line of constant pressure."

Lines of constant heat such as those marked 1200-1190, etc., may be drawn as follows:

The total heat above 493° abs. in dry steam at $C$ has been found to be 1169.6 B.T.U. Let it be required to plot a line of which each point shall represent superheated steam containing 1200 B.T.U. per pound. One point of the line may be found on the constant-pressure line $CD$. The heat at $C$ being 1169.6 B.T.U., it will be necessary to add 30.4 B.T.U. to dry steam in order to produce superheated steam contain-

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* The value of the specific heat used in plotting the curves in the diagram on the back cover of the book is 0.58.

ing 1200 B.T.U. per pound. If $S$ is the specific heat at the pressure represented by $CD$, then the rise of temperature corresponding to the addition of 30.4 B.T.U. per pound may be found from the equation

$$30.4 = S(T_G - T_C).$$

Calling the value of $S$ equal to 0.57 as before,

$$30.4 = 0.57(T_G - 750),$$

or

$$T_G = 803.3 \text{ degs.}$$

This fixes one point of the constant-heat curve for 1200 B.T.U. per pound. A similar method may be followed along all constant-pressure curves for finding the required series of constant-heat curves.

### EXAMPLES IN THE USE OF THE HEAT DIAGRAM.

Let a pound of water be at temperature 600° abs., represented by the point $A$, page 49. It contains sufficient heat above the melting-point of ice to have raised its temperature from that point, or 493°, to its present temperature of 600°, and during that rise in temperature its entropy value has been increased from the arbitrarily assumed zero to the value 0.20. Let heat be added to the water sufficient to raise its temperature to 750° abs. The quantity of heat necessary may be found from the steam-tables by subtracting the heat of the liquid at 600° from that at 750°.

| | |
|---|---|
| Thus, heat of liquid at 750 | =258.6 B.T.U. |
| " " " " 600 | =107.2 B.T.U. |
| Heat involved, represented by the area beneath the curve $AB$, | =151.4 B.T.U. |

Or this might have been found thus:

Temperature range $= 750° - 600° = 150°$.

Mean specific heat of water between the temperatures $= 1.008$.

Heat of liquid

$$= H_L = S(T_B - T_A) = 1.008 \times 150 = 151.4 \text{ B.T.U.}$$

(*a*) If the pound of water were part of the contents of a steam-boiler carrying 57 pds. pressure per sq. inch (corresponding to 750 deg.) and a valve were suddenly opened admitting the water into a large tank in which there was a pressure of only 2.8 pds. abs. (corresponding to 600 deg.) the heat in the water would instantly cause vaporization of the water at the lower pressure and the formation of a great amount of steam. If the valve were opened suddenly enough, the liberation of heat energy caused by the reduction in pressure would occur without transfer of heat to the surroundings, excepting as the latter were disturbed by the external work accompanying the formation of steam. The process would then be adiabatic and represented by the line $BE_1$. The heat available for the formation of steam at the lower temperature and pressure would be measured by the area $ABE_1A$ and would be equal to the heat represented by the area beneath $AB$, down to absolute zero of temperature, minus that represented by the area beneath $AE_1$. Thus, the heat liberated from the water $= H_w = 151.4 - (\text{entropy change from } A \text{ to } E_1) \times 600 = 151.4 - (0.42 - 0.20)600 = 19.4$ B.T.U. per pound.

If the boiler contained 40,000 pds. of water and 450 cu. ft. of steam at 57 lbs. per sq. inch the weight of the steam present would be $450 \div 7.45 = 60$ pounds, and each pound would liberate heat represented by the area $ABCEA$, or 202 B.T.U., approximately. The total heat liberated by 60 pounds steam would be $60 \times 202 = 12,120$ B.T.U., or 9,430,000 ft.-pds.

The heat liberated by the water would be $40,000 \times 19.4 = 776,000$ B.T.U., or about 600,000,000 foot-pounds of energy.

The boiler pressure assumed in the present example is from one third to one quarter of that commonly carried on boilers of the Scotch Marine type, but it gives a means of grasping the reason for the disastrous effects of a boiler explosion, where the contents of a boiler are allowed to expand instantly to a lower pressure and temperature. It is evident, also, that the destructive power is almost wholly due to the large quantity of water carried in the boiler and not to the steam present at any one time.

(*b*) *Formation and adiabatic expansion of steam.*—If, instead of being allowed to expand from $B$ to $E_1$, the water were evaporated into steam by the addition of heat along the isothermal $BC$, the heat necessary to entirely evaporate a pound of water would be represented by the area beneath the line $BC$, and extending down to the absolute zero of temperature. The total amount of heat contained by the pound of steam, above 493 degrees absolute, would then be the sum of the heat of the liquid and that of vaporization, or $258.6 + (\text{entropy change from } B \text{ to } C \times 750) = \text{approximately } 258.6 + 1.215 \times 750 = 1169.6$ B.T.U.

The cycle under consideration, however, does not begin at 493 degrees absolute, but at 600 degrees, indicated at the point $A$. The heat that has been added to that possessed by the water at $A$ is

$$H_w + H_v = 151.4 + 1.215 \times 750 = 1062.4 \text{ B.T.U.}$$

This is the heat represented by the area beneath the broken line $ABC$ down to absolute zero of temperature.

If, now, adiabatic expansion should occur down the line $CE$, that is to the lowest available pressure and temperature (that at $E$), the heat available for transformation into kinetic energy would be that represented by the area $ABCE$, or

$$19.4 + \text{entropy change along } BC \times (750 - 600)$$
$$= 19.4 + 1.215 \times 150 = 201.7 \text{ B.T.U.}$$

The heat rejected into the condenser would be that beneath the line $AE$, or

Heat rejected = change of entropy along $AE \times 600 = (1.64 - 0.20) \times 600 = 364$ B.T.U., approximately.

The efficiency of the cycle is

$$\frac{\text{Heat utilized}}{\text{Heat supplied}} = \frac{201.7}{1062} = 0.19.$$

The working substance, after expansion as steam followed by condensation, would again be in the state of water, represented by the point $A$, and would be ready to be heated again to the boiling-point, evaporated, and carried through the cycle of operations as before.

If not enough heat had been added to completely evaporate the water into dry steam, the state point would have reached some such point as $H$, and the quality of the steam, or percentage of dry steam present, would have been equal to entropy $BH \div$ entropy $BC$. On the diagram the quality and also the heat contents above 493 degrees absolute can be found by interpolation between the quality curves and the total heat curves respectively.

(*c*) *Formation and expansion of superheated steam.*—After dry steam has been formed, thereby bringing the state point to $C$, the addition of further heat results in "superheated steam," or steam having a higher temperature than that at which it was generated, and corresponding to the pressure at which it exists.

The curves for constant-pressure and constant-heat contents for superheated steam have been explained.

Suppose heat to have been added to the dry steam at $C$ until the temperature rises to that at $D$, or 920 degrees absolute. It has been shown that if the specific heat of superheated steam at the pressure under consideration is 0.57, the heat necessary to raise the temperature of dry steam from 750 to 920 degrees will be

$$H_s = 0.57(920 - 750) = 97 \text{ B.T.U.}$$

The total heat above 493 absolute in the pound of steam at $D$ is approximately $258+911+97=1266$ B.T.U., and this is represented by the area beneath the broken line $XABCD$ down to absolute zero of temperature.

Suppose the pound of steam to expand adiabatically from $D$ to the condenser temperature and pressure at $E'$. The heat in the steam above the starting temperature at $A$ is, approximately,

$$H_T=151+911+97=1159 \text{ B.T.U.},$$

and this is represented by the area beneath $ABCD$ down to absolute zero. But only the heat above the horizontal line $AE'$ is available for transformation into kinetic energy, and this equals

$$1159-(\text{change of entropy along } AE')\times 600$$
$$=1159-1.54\times 600=236 \text{ B.T.U.}$$

The efficiency of the ideal cycle is, then, $236\div 1159=0.204$. It is evident that the efficiency of the ideal cycle is not greatly increased by adding the above amount of superheat to steam of the low pressure assumed in the example. The superheat would, however, decrease the losses by condensation, friction of steam, etc., and so increase the efficiency of the actual cycle.

It is to be noted that the steam would remain superheated during expansion until reaching the point $S$, when it would become just dry and saturated. Below $S$ expansion would cause condensation, and at $E'$ the quality of the steam would be represented by entropy $AE'\div AF$.

(*d*) Suppose superheated steam at $D$, containing 1159 B.T.U. per pound above the starting-point at $A$, to expand along some path such as $DE''$, instead of along the adiabatic $DE'$, but falling finally to the same lower pressure as before (note that the line $FE''$ represents the same pressure as does the horizontal $AF$). The position of the point $E''$ indicates that the steam contains, after expansion to $E''$, 1145 B.T.U. per pound, above

493 degrees absolute. Since the heat of the liquid at $A$ is 107 B.T.U., approximately (from the steam-tables), the heat at $E''$ above that at $A$ is $1145-107=1038$ B.T.U.

The steam now falls in temperature, at the condenser pressure, to the lowest available temperature, that at $F$, and in so doing gives up the heat beneath $E''F$, which equals $1145-1124=21$ B.T.U. The heat at $F$ above that at $A$ then equals $1038-21=1017$.

The total heat above $A$ which was available at $D$ before expansion was 1159 B.T.U. Of this, 1017 B.T.U. are to be rejected, and the heat utilized is $1159-1017=142$ B.T.U.

The efficiency of the cycle is $142 \div 1159 = 0.129$.

The falling off in efficiency is due to the fact that the steam has been prevented from attaining the lower temperature attained after adiabatic expansion, and that no steam has been condensed during the expansion. Thus it contains, at the end of expansion to the lowest available pressure, a very much larger amount of heat than it contained after adiabatic expansion to $E'$, and that larger amount of heat has to be rejected to the condenser. The conditions tending to prevent adiabatic expansion will be taken up in the next chapter.

The temperature-entropy chart at the back of the book forms a graphical steam-table, calculated by means of the principles stated in the foregoing pages.

The curve marked "Pressure and Temperature Curve" renders it possible to find the absolute temperature for any of the absolute pressures at the top of the chart. Having found the temperature corresponding to any pressure the specific volume of dry steam at that temperature may be found from the terminations of the constant-volume lines in the dry-steam line. Thus, let it be required to find the absolute temperature corresponding to 120 pounds absolute pressure. Passing down the line marked 120 at the top of the chart until the pressure-temperature curve is reached, the intersection is at the height corresponding to 802 degrees absolute, as nearly as can be read on the chart. By consulting steam-tables the

figure given is 801.9 degrees. For finding the specific volume (cubic feet per pound of dry saturated steam) the line of 802 degrees intersects the saturation curve at a point lying between the lines of constant volume for 3 and 4 cubic feet. The short lines intersecting the saturation curve mark off quarters of cubic feet in the portion of the chart under consideration. At lower temperatures and greater specific volumes the distances between the volume curves represent greater differences. The intersection giving the specific volume for 802 degrees absolute is just above the line marking 3.75 cubic feet, and interpolation gives about 3.7 cubic feet as the volume required. In the steam-tables the volume is given as 3.71 cubic feet per pound. This is, of course, for dry steam of quality 100 per cent. If it is desired to know the specific volume for any other quality of steam, it is simply necessary to find the intersection of the same temperature line with the quality line desired, and to interpolate between the volume lines for the specific volume. Suppose the specific volume of steam of 120 pounds absolute and 95 per cent quality is desired to be known. Passing to the left from the saturation curve along the line of 802 degrees absolute, until a point is reached half-way between the curves of 90 and 100 per cent quality, the specific volume is found to be 3.5 cubic feet.

While the temperature-entropy form of heat-diagram is most admirably adapted to the graphical illustration of heat changes, and of thermodynamic problems in general, and should be thoroughly studied by the student, the diagram proposed by Dr. Mollier having temperature as ordinates and thermal units as abscissae is more readily used in the solution of such problems as come to the engineer. The Mollier diagram at the back of the book will be found to facilitate the problem work called for in the following chapters.

# CHAPTER IV.

## CALCULATION OF VELOCITY AND WEIGHT OF FLOW.

By means of the principles stated in Chapters II and III, the heat drop accompanying the expansion of steam may be calculated, and from this may be found the steam velocity that would result if all the heat given up during expansion were realized as kinetic energy in the jet of steam. It was shown in Chapter II that if $H_1$ and $H_2$ represent respectively the heat contents of the steam, per pound, before and after expansion through an orifice or a nozzle, the velocity equation may be written

$$\frac{V^2}{2g}=778(H_1-H_2). \quad . \quad . \quad . \quad . \quad . \quad . \qquad (11)$$

The velocity of flow may be calculated from the following equations:

Let $q_1$ and $q_2$ represent the heat of the liquid at the higher and lower temperatures, respectively.

Let $E$ represent entropy changes as marked on Fig. 21 and indicated by the subscripts used with the letter $E$.

Let $H_v$ represent the heat of vaporization present in steam of quality $x=1$.

Let $T_1$ and $T_2$ represent absolute temperatures of dry saturated steam at boiler pressure and exhaust at condenser pressure, respectively.

Let $T_3$ represent the absolute temperature to which the steam is superheated.

For moist steam (of quality $=x$), or dry steam (of quality $x=1$)

$$\frac{V^2}{2g}=778\{q_1-q_2+xH_v-T_2(E_1+xE_v-E_2)\}. \quad (12)$$

For superheated steam, calling the specific heat $S$,

$$\frac{V^2}{2g}=778\left\{q_1-q_2+H_v+S(T_3-T_1)\right.$$
$$\left.-T_2\left(E_1+E_v+S\log_e\frac{T_3}{T_1}-E_2\right)\right\}. \quad (13)$$

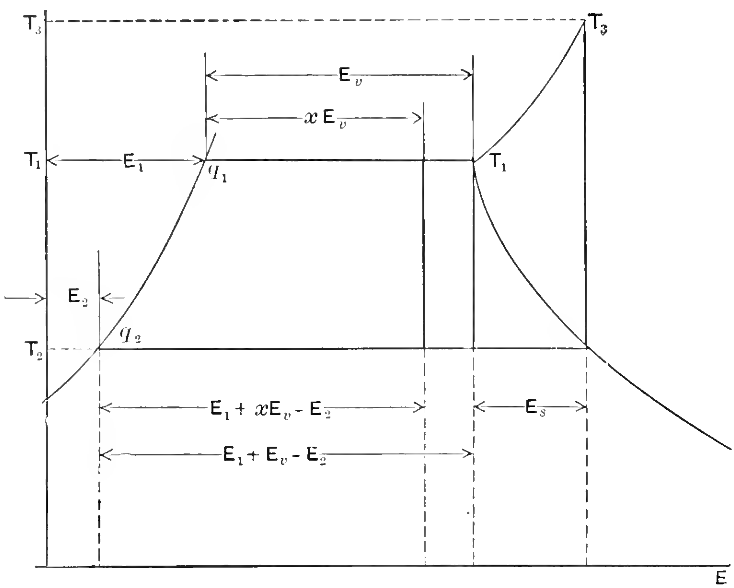



FIG. 21.

Experimental and mathematical investigations indicate in general that the pressure within an orifice through which steam is flowing does not fall below about 0.57 or 0.58 of the initial absolute pressure. It seems that the pressure falls to a value corresponding to the heat conversion that will give to the steam

immediately in the narrowest section of the orifice a velocity about the same as that of the disturbance called "sound," or about 1400 or 1500 feet per second.* Any farther fall in pressure and temperature must take place beyond the narrowest section, but the velocity attained in the narrowest section determines the *weight* of flow, per unit of time, that it is possible for a given initial pressure to produce.

The general explanation of this phenomenon is found in the development of Zeuner's equation given on pages 36 and 37, which indicates that for any fluid flowing through an orifice, the fall of pressure immediately in the orifice is limited to a fraction of the initial pressure depending for its value upon the ratio of the specific heats of the substance at constant pressure and at constant volume, respectively. In the case of steam the pressure falls to that value which gives the maximum possible flow, by weight, and does not fall below this pressure until after the steam leaves the smallest portion of the orifice. The maximum flow from a nozzle leading from a simple orifice may occur when the exit or "back" pressure is higher than $0.57p_1$, as shown on Plates IV, V, and VI; but in these cases the pressure in the orifice is lower than that at exit from the nozzle (see Fig. 53), and the fall of pressure in the throat determines the weight of flow. It is to be noted in this connection that the limiting velocity of 1400 to 1500 feet per second applies to only the narrowest portion of the nozzle, and that farther fall in pressure beyond this point may very considerably increase the velocity of the stream.

Referring to Plates IV, V, and VI, curves No. 2 show experimentally determined weights per second flowing from orifice No. 2, the entrance to which is rounded. Assuming that in each case the orifice pressure is 0.57 of the initial pressure when the maximum weight of steam flows through the orifice, calculations according to the equation on page 62 for moist and for dry steam give the following results:

* The rate of wave propagation depending upon the temperature in the orifice. See "Outflow Phenomena of Steam," Paul Emden. Munich, 1903. R. Oldenbourg.

TABLE No. 1.

| Initial Pressure, Pounds Absolute per Square Inch. $P_1$ | Orifice Pressure. $P_2=0.57P_1$. | Weight of Flow, Pounds per Second. Observed. | Calculated By Equation Given. | Calculated By Napier's Formula. | Calculated Velocity, Feet per Second at Smallest Cross-section of Orifice. |
|---|---|---|---|---|---|
| 132.3 | 75.2 | 0.063 | 0.0629 | 0.0671 | 1470 |
| 117.6 | 67.0 | 0.057 | 0.0572 | 0.0596 | 1495 |
| 102.9 | 58.7 | 0.050 | 0.0500 | 0.0522 | 1490 |

The experiments upon which the above table and the curves on Plates IV, V, and VI are based were made by Professor Gutermuth, of Darmstadt, and were published in the Zeitschrift des Vereines Deutscher Ingenieure, Jan. 16, 1904.

The following specimen calculation shows the method of using the equations on page 62. Taking the conditions in the first case in the above table,

$P_1=132.3$ pds. absolute per sq. inch;
$q_1=$heat of liquid at $P_1=319.3$ B.T.U.
$H_v=$ " " vaporization at $P_1=868.4$ } $q_1+H_v=1188$ B.T.U.;
$E_1+E_v=$specific entropy of steam at $P_1=1.573$;
$P_2=$pressure in the orifice, pds. abs. per sq. inch $=75.2$ pounds;
$q_2=$heat of liqu d at $P_2=277$ B.T.U.;
$T_2=$absolute temperature at $P_2=768°$ F.;
$E_2=$specific entropy of water at $P_2=0.446$;
Specific volume of dry steam at $P_2=5.75$ cu. ft.

From equation (12), for moist or dry steam, on page 62,

$$V^2 \div 2g=778(q_1-q_2+H_v-T_2(E_1+E_v-E_2)),$$

from which $V=1470$ ft. per second.

Calculating the quality of steam after expanding adiabatically from 132.3 to 75.2 pounds absolute, the specific volume at the lower pressure will be $0.965\times5.75=5.55$ cu. ft.

The steam flows through an orifice of 0.0355 in. cross-sectional area at the velocity 1470 f . per second.

Since for steady flow

$$\text{Volume discharged} = \text{area of orifice} \times \text{velocity},$$

the volume flowing per second $= 0.0355 \div 144 \times 1470 = 0.362$ cu. ft., and since each cubic foot weighs $\frac{1}{5.75}$ pounds, the weight flowing per second is $\frac{0.362}{5.75} = 0.0629$ pounds.

**The calculation for the weight of flow through an orifice** may be simplified by an approximation to the area of the heat diagram, as follows:

Assuming that steam of quality $\frac{HN}{HM}$ expands adiabatically along the line $NA$ (Fig. 22), the heat given up is represented by

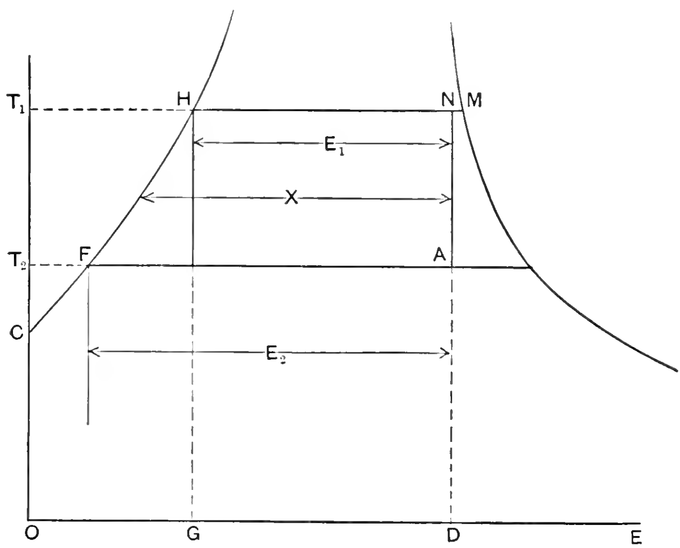



FIG. 22.

the area $FHNAF$ lying between the limits of temperature $T_1$ and $T_2$. This area is equal to the mean width of the area multiplied by the range of temperature, or since $FH$ is nearly

a straight line, the heat causing flow $= X(T_1 - T_2) =$ approximately

$$\frac{(\text{entropy } HN + \text{entropy } FA)}{2}(T_1 - T_2).$$

Taking the data in the second line of the table on page 64,

$P_1 = 117.6$ pds. abs.
$T_1 = 800$ degs. abs.
$P_2 =$ pressure in orifice, or $0.57P_1 = 67$ pds.
$T_2 = 761$ degs. abs.

Assume that the quality of the entering steam is 100% or that $N$ coincides with $M$, then

$$\begin{array}{lr} \text{Entropy } HN & = 1.093 \\ \text{Entropy } FA = 0.053 + 1.093 & = 1.146 \\ \hline \text{Entropy } HN + FA & = 2.239 \end{array}$$

$\frac{2.239}{2} = 1.12$ entropy corresponding to mean ordinate.

$$800 - 761 = 39 \text{ degs.}$$

$$39 \times 1.12 = 43.6 \text{ B.T.U.}$$

$$\text{Velocity} = \sqrt{778 \times 43.6 \times 64.4} = 1480 \text{ ft. per second.}$$

The velocity calculated on page 64 is 1495 feet per second. The specific volume of steam at the orifice pressure of 67 pds. is 6.4 cu. ft. Cross-sectional area is 0.0355 sq. inch.

The weight flowing per second is then

$$\frac{0.0355 \times 1480}{144 \times 6.4} = 0.057 \text{ pound.}$$

Let area of orifice in sq. inches be called $A$;

specific volume (cu. ft. per pound) of steam after expanding to $P_2$ $(= 0.57P_1)$ be called $v_2$;

entropy values be designated by letter $E$ with subscripts, that is, as $E_1$ and $E_2$, Fig. 22.

temperatures corresponding to $P_1$ and $0.57\ P_1$ be called $T_1$ and $T_2$ respectively.

The weight flowing per second =

$$W=\frac{A}{v_2\times 144}\sqrt{778\times 64.4\left\{\frac{E_1+E_2}{2}(T_1-T_2)\right\}}$$

$$=\frac{1.1A}{v_2}\sqrt{(E_1+E_2)(T_1-T_2)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

The formula may be extended so as to include cases in which superheated steam is used, by adding to the expression under the radical the equivalent of the superheat in the steam per pound.

The volume $v_2$ after expansion to $0.57P_1$ will be very nearly 96.5% of the specific volume at the pressure $0.57P_1$. This may be verified by means of the heat diagram by finding the quality of steam after the expansion stated.

**Calculation of Rate of Flow and of Reaction against the Outflow Vessel.**—If the reaction due to a jet delivering a given weight of substance per unit of time be known the velocity of the jet can be computed.

The velocity of a jet is produced by a force urging the substance onward, and the work done by this force is the equivalent of the heat given up by the steam during its fall in pressure and temperature as it flows through the orifice, or nozzle.

**Nature of the Reaction.**—A jet in flowing from an orifice in a chamber suspended by a flexible tube as in Fig. 23, causes the chamber from which the jet flows to move in a direction opposite to that of the flow of steam, and to assume some new position, as indicated by the dotted lines. While the force holding the chamber in this new position is the equivalent of the force urging the jet onward, and may therefore be used as such in computing the velocity of the jet, the true nature of the influence producing the reaction is not brought out by such an explanation.

If a force could be conceived to act back through the stream and thus push the chamber into the new position, it would be necessary to conceive also of a point of application of the force

to the object moved—that is, to the chamber from which the jet flows. If the force were applied to the steam within the chamber, the unit pressure within would be increased. This is contrary to observation, and, besides, such an increase in pressure would apply to all sides of the chamber, and no un-

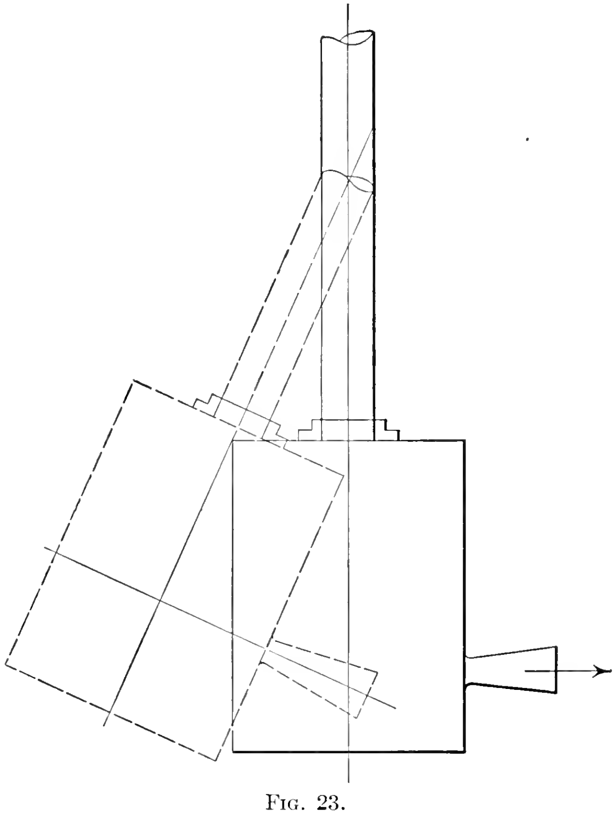


FIG. 23.

balanced forces would arise to cause displacement of the chamber as a whole.

It would be difficult to conceive of a force acting in a direction opposite to that of the flow of steam, and being applied to the edges of the orifice in such a way as to affect the position of the chamber.

Without speculating further, the removal of pressure at the entrance to the orifice allows the steam about the entrance to expand in volume, to fall in pressure and temperature, and to be forced through the orifice by that part of the intrinsic energy of the steam itself which is given up during the expansion, and converted into the kinetic energy of flow. The diminution of pressure about the entrance to the orifice while the pressure on the other surfaces of the interior of the chamber remains the same as before results in an unbalanced force within the vessel, causing displacement of the vessel as a whole. Equilibrium is restored only when the elasticity of the supporting tube causes a force sufficient to balance the resultant of the internal pressures.*

If a conically divergent nozzle of suitable proportions be added to the orifice on the side of the chamber, the expansion of the steam after it leaves the orifice may, with certain initial pressures, result in a higher velocity of flow in a given direction than occurs after expansion through a simple orifice. If the steam, before leaving the large end of the nozzle, expands down to the external pressure at the exit from the nozzle, then the velocity of flow will be as great as it is possible to attain with the pressures involved and the particular nozzle in question.

The question arises, since an increase in velocity must be accompanied by an increased reaction, where does the additional unbalanced force find its point of application? Assuming that for a given nozzle and given initial pressure definite orifice conditions exist as to pressure and rate of flow, the conditions of expansion in the part of the nozzle beyond the orifice may be supposed to not affect the orifice conditions. Taking two sections indefinitely near to each other, at which pressures $p$ and $(p-dp)$ exist, a pressure $p'$ acts in the direction $AC$, normal to the nozzle surface, upon each elementary area, and may be resolved into two components, $AB$ and $BC$, perpendicular and parallel, respectively, to the direction of flow. If the nozzle sides make an angle $\alpha$ with the direction of flow

* Neglecting the weight of the parts.

the components along and perpendicular, respectively, to that direction are (Fig. 24):

$$BC = p' \sin \alpha,$$
$$AB = p' \cos \alpha.$$

If the rate of pressure fall along the nozzle be assumed, the integration of the above expressions over the interior surface of the nozzle will give values for the components $AB$ and $BC$, the latter representing the reaction against the nozzle. Further analysis would not assist in the following application of the reaction principle to problems in the flow of steam, since the pressures in the nozzle vary in a complex manner;

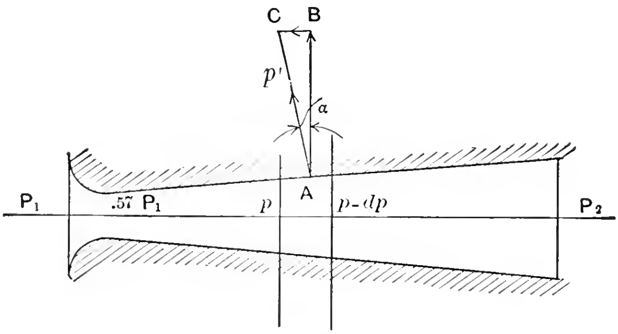



FIG. 24.

but the above indicates the general character of the forces involved. It is evident that the reaction accompanying flow through a straight nozzle or pipe would not differ from that through a simple orifice, except that the rate of flow would be affected by the friction caused by the nozzle walls.

The development of equation 14 shows that the maximum possible velocity due to adiabatic expansion from $P_1$ to $P_2$ is, approximately,

$$V = \sqrt{50103 \frac{(E_1 + E_2)}{2} (T_1 - T_2)}$$
$$= 158 \sqrt{(E_1 + E_2)(T_1 - T_2)}, \quad . \quad . \quad . \quad (15)$$

where $E_1$ and $E_2$ represent entropy changes, as stated on page 66.

If the expansion is that occurring in an orifice, the range of pressures is between $P_1$ and $0.57P_1$, and at the higher pressures, that is between 200 pounds and 100 pounds, the value of the expression under the radical is from 9.60 to 9.70. Below 100 pounds the value is from 9.0 to 9.4. Taking an average value of 9.5, the limiting velocity in the plane of an orifice is, approximately,

$$158 \times 9.5 = 1500 \text{ ft. per sec.}$$

The weight of flow may be calculated by using the following formula:

$$W = \frac{AV_2}{v_2 \times 144},$$

where $A$ = area of orifice in square inches;
$V_2 = 1450$ for initial pressures below 100 pds. abs.;
$V_2 = 1520$ for initial pressures above 100 pds. abs.;
$v_2$ = cubic ft. per pound at pres. of $P_2 = 0.57P_1$.

For example,

Let $P_1 = 155$ pounds per sq. inch absolute. Then $P_2 = 0.57 \times 160 = 88$ pounds.

$v_2 = 4.96$.

Let $A = 0.0275$ sq. in.

$$\text{Weight per second} = \frac{0.0275 \times 1520}{144 \times 4.96} = 0.0585.$$

This result may be compared with the result for 155 pounds pressure on page 92.

A more satisfactory formula, however, is derived from the velocity as given on the preceding page, as follows:

$$V = 158\sqrt{(E_1 + E_2)(T_1 - T_2)},$$

$$W = \frac{158A}{144v_2}\sqrt{(E_1 + E_2)(T_1 - T_2)},$$

$$= \frac{1.1A}{v_2}\sqrt{(E_1 + E_2)(T_1 - T_2)}.$$

This will be found to give results agreeing very closely with the actual weight of flow from orifices with rounded entrance.

The statement is frequently made and seems to have been largely accepted, that steam flowing through a simple orifice cannot attain a velocity greater than about 1500 feet per second. This is probably true for the position immediately at the smallest section through which the steam passes, but it should not therefrom be concluded that the total kinetic energy possessed by a jet from an orifice is limited to the amount corresponding to that velocity. It seems that in flowing through a simple orifice steam gives up energy until it attains a velocity corresponding to about that stated, but that after that state of activity has been reached, further acceleration does not occur until the narrowest section has been passed. As soon as the steam reaches a point just beyond that section, however, it is free to expand to the pressure of the medium into which the orifice leads. The jet issues in a well-formed stream in a given direction, and as it falls in temperature the heat liberated tends to further accelerate the jet in the direction of motion. If there is no directing nozzle beyond the orifice, however, the jet begins to spread soon after leaving the orifice, and hence its kinetic energy is given up in directions other than that of the original jet. The same amount of energy is given up by a jet from an orifice as from an expanding nozzle, but the latter, if properly proportioned, serves to contain the steam during expansion so that the maximum possible velocity in a given direction is obtained with little vibration of the atmosphere and consequent loss of energy.

The experimental work discussed in Ch. VI indicates that much higher velocities than ordinarily supposed are possible by the use of orifices, and it has been found in building certain turbines of the impulse type that fully as good, if not better, results are obtained in the lower stages of turbines by the use of orifices instead of nozzles. The latter are especially suited to pressures above 70 or 80 pounds absolute.

In the ideal case, used for predicting results to be expected,

the following steps may be taken towards calculating the weight of flow and velocity.

(*a*) Find the weight of flow caused by the fall in pressure in the orifice to $0.57P_1$, as in the equation

$$W=\frac{1.1A}{v_2}\sqrt{(E_1+E_2)(T_1-T_2)}.$$

(*b*) Find the velocity corresponding to the heat given up during drop in pressure to that existing at the exit of the orifice or nozzle, or $P_3$, from equation (15):

$$V=158\sqrt{(E_1+E_3)(T_1-T_3)},$$

where $E_3$ is the entropy change (marked $E_2$ on the diagram Fig. 22), and $T_3$ is the corresponding absolute temperature.

(*c*) Correct these by experimentally determined coefficients for friction and other losses, as will be explained in the following chapter.

(*d*) If the weight of steam flowing through the passageway per unit of time has been determined experimentally, or if the reaction has been so found, it may be useful to employ these values for calculating the actual velocity.

The reaction in pounds has been shown to equal the weight of flow per second times velocity in feet per second divided by $g$ ($=32.2$). The equation for calculating the reaction may be written

$$R=\frac{WV}{g}=\frac{158}{32.2}\{(E_1+E_3)(T_1-T_3)\}^{\frac{1}{2}}\times\frac{1.1A}{v_2}\{(E_1+E_2)(T_1-T_2)\}^{\frac{1}{2}}$$

$$=\frac{5.4A}{v_2}\{(E_1+E_3)(E_1+E_2)(T_1-T_3)(T_1-T_2)\}^{\frac{1}{2}}. \qquad (16)$$

In the above, $A$ = area of least cross-section of passage, in square inches.

$v_2$ = specific volume of steam at $0.57P_1$, in cubic feet.

Values of $v$, $E$, and $T$ may all be taken from the heat diagram directly, with sufficient accuracy for engineering purposes.

An equation for calculating the reaction of a jet of steam flowing into the atmosphere was developed about the time when Mr. George Wilson's experiments were made (1872), and although the equation must be regarded as empirical, it expresses with remarkable closeness the results that have been obtained as to the reaction of steam-jets discharging into the atmosphere. The reasoning made use of in developing the equation was somewhat as follows:

If steam be allowed to expand behind a piston in a cylinder from $P_1$ to $0.57P_1$, adiabatically, the mean effective pres ure will be about $0.33P_1$. If a stream capable of exerting this mean pressure were allowed to flow through an orifice, it would be able, according to the principles governing the impulse of jets of fluid, to exert an impulsive pressure, and therefore a reaction, of twice the pressure corresponding to its static head, or of $0.66P_1$. Besides this pressure the reaction would be increased by the addition of the pressure in the orifice, or $0.57P_1$, but as the flow is into the atmosphere, and $P_1$ is in pounds absolute, the atmospheric pressure must be subtracted. The expression for the reaction then becomes

$$R = P_1(0.66+0.57)-14.7=1.23P_1-14.7 \text{ lbs. per sq. in. of orifice.}$$

The following table * shows the degree of approximation to experimentally determined reactions which can be attained by use of the equation. The experiments were made by Mr. George Wilson with the apparatus shown on page 140.

Further calculations by means of the formula just developed are given in Chapter VI. If it be attempted to apply the formula to cases of discharge into a condenser maintaining conditions of partial vacuum, it will appear that the results are not in accordance with calculations made on the basis of heat given up. The maximum velocity of flow of a jet discharging into a perfect vacuum would be, from the formula, that corresponding to a reaction of $1.23P_1$. For steam of an

* Proceedings of Engineers and Shipbuilders of Scotland, 1874–5

| Absolute Pressure. Pounds per Square Inch. | Reaction, Orifice 1.0956 Sq. Ins. Area, by Experiment. | Reaction Calculated. | $\frac{\text{Calculated}}{\text{Experimental}}$. |
|---|---|---|---|
| 16.49 | 3.54 | 3.63 | 1.025 |
| 18.10 | 6.52 | 6.78 | 1.040 |
| 19.98 | 9.86 | 10.05 | 1.019 |
| 23.10 | 14.75 | 14.92 | 1.011 |
| 24.85 | 17.27 | 17.37 | 1.005 |
| 25.60 | 18.32 | 18.10 | 0.988 |
| 27.30 | 20.84 | 20.68 | 0.992 |
| 39.40 | 38.00 | 37.00 | 0.973 |
| 54.40 | 59.00 | 59.00 | 1.000 |
| 73.20 | 85.11 | 82.60 | 0.970 |
| | Reaction, Orifice 0.4869 Sq. In. Area. | | |
| 22.80 | 8.10 | 8.21 | 1.013 |
| 42.20 | 18.22 | 18.14 | 0.995 |
| 65.40 | 32.50 | 32.04 | 0.986 |
| 77.00 | 39.55 | 38.51 | 0.973 |
| 84.90 | 44.00 | 43.72 | 0.994 |
| 93.00 | 48.50 | 48.58 | 1.002 |
| 112.70 | 58.30 | 60.38 | 1.034 |
| 55.70 | 26.67 | 26.20 | 0.983 |
| 84.70 | 44.30 | 43.56 | 0.983 |
| 113.70 | 59.60 | 60.90 | 1.022 |
| | Sum. . . . . . . . | . . . . . . . . . . . . . . . . . | . . . 20.008 |
| | Average of 20 | experiments. . . . . | . . . 1.0004 |

initial pressure of 160 pounds absolute per sq. in., discharging into a vacuum of 28 ins. through an orifice of 0.25 sq. in. cross-sectional area, the maximum weight of flow per second would be

$$W = \frac{1.1 \times 0.25}{4.7}\sqrt{2.15 \times 42} = 0.55 \text{ pds.}$$

The reaction would be, by the above equation,

$$R = (1.23 \times 160 - 1.0)0.25 = 49 \text{ pounds,}$$

and the velocity

$$V = \frac{Rg}{W} = \frac{49 \times 32.2}{0.55} = 2870 \text{ ft. per second.}$$

This result may be compared with that obtained by use of

the approximation to the fundamental equation for velocity, eq. (15). If complete expansion occurs, in a suitable nozzle,

$$V = 158\sqrt{(E_1 + E_3)(T_1 - T_3)}$$
$$= 158 \times 25.5 = 4030 \text{ ft. per second.}$$

It will be shown in the following chapter how calculations made by the last used equation may be modified by a suitable coefficient for friction losses in the nozzle or orifice, so as to predict results to be expected in practice.

# CHAPTER V.

## VELOCITY AS AFFECTED BY FRICTIONAL RESISTANCES.

REFERRING to Fig. 25, let a pound of steam be at pressure $p_1$ and volume $v_1$, and let its adiabatic expansion be indicated by curve $p_1v_1 - p_2v_2$. At $p_2v_2$ partial condensation of the pound

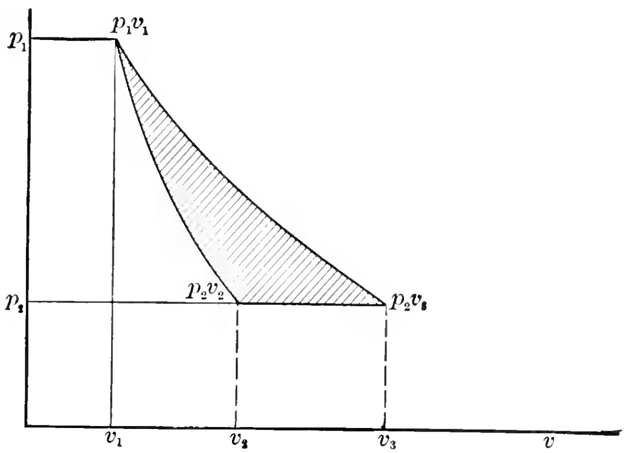



FIG. 25.

of steam has occurred, and there exists a volume $v_2$ of steam, and a certain amount of water, the steam and water together weighing one pound. If the steam contained at $p_1v_1$ the heat $H_1$, and contains at $p_2v_2$ the heat $H_2$, the increase of velocity of the steam that could occur, due to the fall from $p_1v_1$ to $p_2v_2$, is

$$V_2 = \{64.4 \times 778 \times (H_1 - H_2)\}^{\frac{1}{2}}.$$

Now let the pound of steam expand from the same initial condition $p_1v_1$ to $p_2v_3$, in which $v_3$ is greater than $v_2$. Since the final pressure $p_2$ is the same for both cases of expansion, the steam at the condition of greater volume per pound, $v_3$, is more nearly dry than at $v_2$. This means that after expansion to $v_3$ the steam possesses more energy than after expansion to $v_2$, or, in other words, it has given up less of its energy than was given up by expanding adiabatically. Having reached the lowest available pressure and temperature at $p_2$, the steam cannot give up any further energy, because it cannot fall any further in temperature. The shaded area (Fig. 25) represents the difference between the energy (in foot-pounds) given up by the steam in the two cases. Let the quantity of heat remaining in the steam at $p_2v_3$ be $H_2'$. This is greater than $H_2$ because less condensation has occurred during the fall from $p_1v_1$ than occurred during adiabatic expansion.

The velocity of the steam after falling to $p_2v_3$ is

$$V_2' = \{64.4 \times 778 \times (H_1 - H_2')\}^{\frac{1}{2}}.$$

The velocity after adiabatic expansion to $p_2v_2$ is

$$V_2 = \{64.4 \times 778 \times (H_1 - H_2)\}^{\frac{1}{2}}.$$

The difference between the squares of the velocities, or the loss of energy, is evidently represented by

$$V_2^2 - V_2'^2 = V_L^2 = \{64.4 \times 778 \times (H_2' - H_2)\}.$$

Remembering that the quantity of steam involved is one pound, the loss of energy is

$$\frac{V_L^2}{2g} = 778(H_2' - H_2).$$

Let $N$, Fig. 26, represent the initial condition of the pound of steam (at $p_1v_1$ in the pressure-volume diagram), and let expansion occur adiabatically along $NA$ to the temperature corresponding to $p_2$. The amount of steam present at $A$ will be $FA \div FL$ pounds, and the amount of water will be $1 - FA \div FL$ pounds. The quality of the steam will therefore be $x = FA \div FL$. If expansion occurs in a passage which opposes

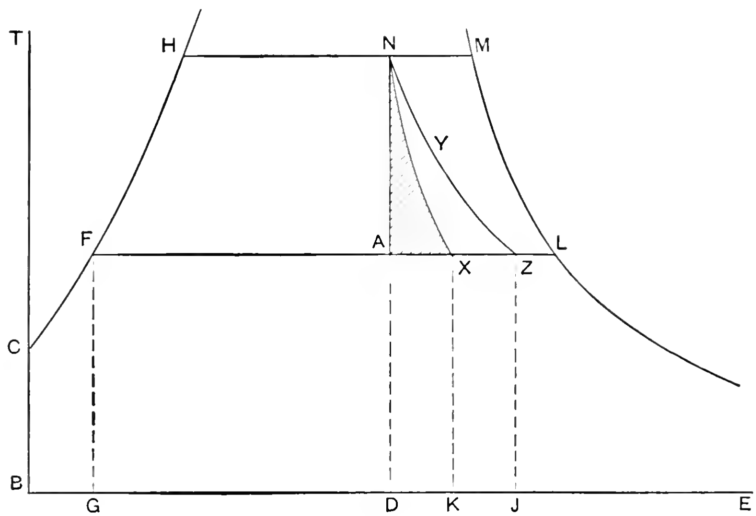



FIG. 26.

frictional resistance to the flow, the steam gives up part of its energy to overcome the resistance, and the work thus done appears as heat in the walls of the passageway, or in the particles of the steam itself. Each indefinitely small drop in temperature is accompanied by this giving up of heat to the surroundings of the steam, and the surroundings give back heat to the steam as soon as the latter falls below the temperature to which the surroundings have been heated. This giving back of heat to the steam re-evaporates the water of condensation resulting from adiabatic expansion and raises the quality of the steam so that expansion occurs along some such line as $NX$. If expansion occurs through a small hole into a comparatively

large chamber, as in a throttling calorimeter, the final velocity of the steam is negligibly small, and the work of friction is all spent in increasing the internal energy of the steam during its fall in temperature. Thus, as practically no heat escapes, the expansion follows the constant heat curve $Y$. At any lower temperature, as at $FL$, the quantity of heat present is the same as was present at $N$, but no external work has been done; and if $FL$ is at the lowest available temperature, the whole of the heat must be rejected and cannot be usefully employed. The case is like that of the water in the tail-race of a mill—it can fall no farther and hence can give up no more energy, although the mass of water present is the same as it was as it flowed in the penstock. The total heat above the starting-point $F$ in a pound of steam at condition $N$, Fig. 25, is

$$H_1 = \text{area } GFHNADG.$$

If unresisted adiabatic expansion occurs along $NA$, the quantity of heat usefully employed in giving velocity to the steam will be that represented by area $FHNAF$.

The heat rejected along the line $AF$ of lowest available temperature will be

$$H_2 = \text{area } GFADG.$$

If the work of friction in the nozzle should be sufficient to cause the steam to fall in temperature along the constant heat curve $Y$, the whole of the heat available at $N$ would exist in the steam after falling to $Z$, and would be rejected along the line $ZF$. The heat so rejected would be

$$H_2' = \text{area } GFZJG,$$

which equals $H_1$, the original heat in the steam at $N$. The total amount of heat available at $N$ would thus fall in tempera-

ture without doing any work towards increasing its own velocity—that is, the velocity at $Z$ being $V_2'$,

$$V_2' = \{64.4 \times 778(H_1 - H_2')\}^{\frac{1}{2}} = 0,$$

since $H_1 = H_2'$.

The work of friction is represented by the area $ANZA$. It is obvious that the work of friction causes the entropy of the steam at its lowest temperature to be greater than it would be if adiabatic expansion occurred from $N$ to $A$. The heat rejected is therefore made greater by an amount represented by the area $ADJZA$, which represents the actual loss of kinetic energy due to friction. The work of friction represented by area $ANZ$ is all returned to the steam, and serves to increase its dryness fraction, but in doing so it decreases the amount of energy the steam is capable of giving up towards increasing its own velocity.

*Example.*

Let the initial pressure at $N = 150.0$ pds. sq. in. $= p_1$;
" " final " " $Z = 1.5$ " " " $= p_2$;
" " quality of steam at $N = 0.90$;
" " steam fall in pressure along the constant heat curve $Y$.

Heat of liquid at 150 pds. abs. $= 330$ B.T.U.
" " vaporization at 150 pds. abs. $= 861$ B.T.U.
$0.90 \times 861 + 330 = 1105$ B.T.U. total heat per pd. at $N$.

Since the heat at $Z$ is to be also 1105 B.T.U. and the total heat of saturated steam at $L$ is 1117 B.T.U., the quality at $Z$ may be found as follows:

Heat of liquid at $F = 84.1$ B.T.U.
Quality at $Z = (1105 - 84.1) \div (1117 - 84.1)$

$$= \frac{\text{entropy } FZ}{\text{entropy } FL} = 0.987.$$

Entropy of vaporization at 1.5 pds. pres. $= 1.790 =$ entropy $FL$.

Entropy $FZ = 1.79 \times 0.987 = 1.766$.

$$\left.\begin{array}{llrl} \text{Heat beneath } FH = 330 - 84 = & & 246 & \text{B.T.U.} \\ \quad '' \qquad '' \quad HN = 0.9 \times 861 = & & 775 & \quad '' \\ & & \overline{\phantom{0000}} & \quad '' \\ & & 1021 & \quad '' \end{array}\right\} = H_1 = \text{heat in steam at initial conditions.}$$

Heat beneath $FZ =$ entropy $FZ \times$ abs. temp. of steam at 1.5 pds. pres.

$= 1.77 \times 577 = 1021$ B.T.U.

$=$ heat in steam at final condition at $Z$

$= H_2'$.

It is evident that $H_1 - H_2' = 0$ and therefore that no velocity would result from fall of temperature along the curve $Y$. It is to be noticed that in the above example the heat represented by area $ADJZA$ equals that by $FHNAF$, since $GFHNDG$ equals $GJZFG$, and $GDAFG$ is common to both areas. Thus the initial available heat just equals the loss of heat caused by the steam following the curve of constant heat.

In general the steam in a nozzle expands according to some such curve as $NX$ between $NA$ and $NZ$, and the shaded area $NAXN$ represents the friction work, while $AXKDA$ represents the loss of energy due to the resistance. Since the friction work is all returned to the steam as heat it is not necessary to determine its value, but the *loss of energy due to the frictional resistance* is one of the most important items connected with steam-turbine calculations.

Let $H_1 =$ heat in entering steam, as defined on p. 80;

$H_2 =$ heat rejected after adiabatic expansion to the lower pressure $p_2$;

$H_v =$ heat of vaporization of dry saturated steam at pressure $p_2$.

If the steam falls in pressure adiabatically, and without

frictional resistance, the heat given up is $H_1 - H_2$ and the velocity developed by the steam-jet is

$$V = \{64.4 \times 778 \times (H_1 - H_2)\}^{\frac{1}{2}}.$$

If $y$ one-hundredths of the heat $H_1 - H_2$ is lost, due to the frictional resistance corresponding to fall down the curve $XX$, Fig. 26, the heat given up will be $(1-y)(H_1 - H_2)$ and the resulting velocity will be

$$V = \{64.4 \times 778 \times (1-y)(H_1 - H_2)\}^{\frac{1}{2}}. \quad . \quad . \quad . \quad (17)$$

The quality of the steam after expanding to $p_2$ against the resistance will be higher than after adiabatic expansion by an amount represented by $AX \div FL$, Fig. 26. This ratio is the same as the ratio between the quantities of heat beneath $AX$ and $FL$ respectively. But the loss of heat, $y(H_1 - H_2)$, is equal to the heat represented by the area beneath $AX$, and the heat beneath $FL$ is equal to the heat of vaporization, $H_v$, of steam at $p_2$. Therefore the increase of quality of the steam, due to the resistance, is

$$x'' = y(H_1 - H_2) \div H_v. \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

The quality at $X$, Fig. 26, is the sum of the per cent of steam at $A$ and the percentage represented by the above expression. Knowing the weight of steam flowing through a passage per unit of time, the volume may be determined from the quality of the steam. Knowing the volume and the velocity the proper cross-sectional area for the passage may be determined.

*Example.*

Let the initial pressure be 150.0 pds. per sq. in. abs. $= p_1$;
" " final " " 1.5 " " " " " $= p_2$;
" " loss of energy in the passage be 15% or $y = 0.15$;
" " initial quality of steam be 0.98.

Then $H_1 = 1090$ B.T.U. = heat above point $F$, Fig. 26.

Entropy at $N$ (Fig. 26) = 1.543.

Entropy $FA$ = 1.543 − entropy $BG$ = 1.543 − 0.157 = 1.386.

$H_2$ = entropy $FA \times$ abs. temp. corresponding to $p_2 = 1.386 \times 577 = 800$ B.T.U.

Velocity $V = \{64.4 \times 778 \times 0.85(1090-800)\}^{\frac{1}{2}} = 3500$ ft. per sec.

The quality of steam at $A$ would be

$$x' = \text{Entropy } FA \div \text{entropy } FL = 1.386 \div 1.791 = 0.774, \text{ approx.}$$

This quality is increased by the amount

$$x'' = y(H_1 - H_2) \div H_v = 0.15 \times 290 \div 1033 = 0.042,$$

or the quality at $X$ is $x' + x'' = 0.774 + 0.042 = 0.816$.

The specific volume of steam at $p_2$, or 1.5 pds. abs., is 227 cu. ft. Neglecting the volume of the water of condensation, the volume per pound of the steam in the present example is

$$227 \times 0.816 = 185 \text{ cu. ft.}$$

In any conduit or passage, if a steady flow of fluid takes place, the volume flowing per second is

$$Q = AV,$$

where $A$ is the area of cross-section of the passage and $V$ is the velocity. If $Q$ is in cu. ft., then $A$ should be in square ft. and $V$ in ft. per second. If the passage varies in cross-section to $A_1$ and the quantity $Q$ remains the same, then $Q = A_1V_1$.

In general, for steady flow the equation may be written

$$Q = AV = A_1V_1 = A_2V_2, \text{ etc.}$$

If the volume varies, then for a given area of cross-section the velocity will vary. In the present example, suppose 0.25 pd. steam flows through an expanding nozzle and reaches at the large end a velocity of 3500 ft. per second, as found above, corresponding to a pressure of 1.5 pds. abs. per sq. in.

The volume per pd. has been found to be 185 cu. ft., or

the vol. flowing per second is $0.25 \times 185 = 46.2$ cu. ft. It is required to find the cross-sectional area of the nozzle at the large end.

$$Q = 46.2 \text{ cu. ft. per sec.}$$
$$V = 3500 \text{ ft. per sec.}$$
$$A = Q \div V = 46.2 \div 3500 = 0.0132 \text{ sq. ft.}$$

or $$0.0132 \times 144 = 1.9 \text{ sq. inches.}$$

*Problem.*—Find the smallest cross-section of a conically divergent nozzle for carrying out the expansion indicated in the above problem, and find three intermediate cross-sections, where the pressures will be 75, 50, and 25 pds. abs. respectively. Make the nozzle 8 inches long and sketch it on cross-section paper.

CALCULATIONS.

| Calculations for $p_2 = 1.5$ Pounds Absolute. | $p_2 = 1.5$ Pds. Abs. | $p_2 = 7.5$ Pds. Abs. | $p_2 = 15$ Pds. Abs. | $p_2 = 25$ Pds. Abs. | $p_2 = 50$ Pds. Abs. | $p_2 = 75$ Pds. Abs. |
|---|---|---|---|---|---|---|
| $H_1 = (0.98 \times 861) + (330 - 84) =$ (B.T.U.) | 1090 | 1025 | 992 | 965 | 924 | 897 |
| $E_N$ | 1.543 | 1.543 | 1.543 | 1.543 | 1.543 | 1.543 |
| $E_{BG}$ | 0.157 | 0.264 | 0.314 | 0.354 | 0.411 | 0.446 |
| $E_{FA} = E_N - E_{BG}$ | 1.386 | 1.279 | 1.229 | 1.199 | 1.132 | 1.097 |
| $H_2 =$ abs. temp. $T_2 \times E_{FA}$ (B.T.U.) | 800 | 820 | 829 | 840 | 840 | 842 |
| $H_1 - H_2$ (B.T.U.) | 290 | 205 | 163 | 125 | 84 | 55 |
| $V = \sqrt{[64.4 \times 778 \times (1.00 - 0.15)(H_1 - H_2)]}$ (ft. per sec.) | 3500 | 2960 | 2640 | 2310 | 1890 | 1530 |
| $E_{FL} =$ entropy of vaporization at $p_2$ | 1.791 | 1.542 | 1.432 | 1.350 | 1.237 | 1.169 |
| Quality at $A = E_{FA} \div E_{FL} = \frac{1.386}{1.791}$ | 0.774 | 0.829 | 0.856 | 0.888 | 0.915 | 0.939 |
| Heat of vaporization at $p_2 = H_v$ (B.T.U.) | 1033 | 988 | 965 | 946 | 917 | 898 |
| Increase in quality along $AX = y(H_1 - H_2) \div H_v$ | 0.042 | 0.0311 | 0.0253 | 0.0198 | 0.0137 | 0.0092 |
| Quality at $X = 0.774 + 0.042$ | 0.816 | 0.860 | 0.881 | 0.908 | 0.927 | 0.948 |
| Sp. vol. dry steam at $p_2$ (cu. ft.) | 227 | 50.0 | 26.1 | 16.1 | 8.41 | 5.75 |
| Vol. per pd. of wet steam, $227 \times 0.816 = v_2$ | 185 | 43.0 | 23.0 | 14.6 | 7.81 | 5.28 |
| Vol. per sec. $= 0.25 \times 185 = Q$ | 46.2 | 10.75 | 5.75 | 3.65 | 1.95 | 1.32 |
| Area (sq. in.), cross-section of nozzle $= Q \div V = \frac{46.2 \times 144}{3500}$ | 1.9 | 0.523 | 0.313 | 0.227 | 0.148 | 0.124 |
| Diameter of nozzle, ins. | 1.56 | 0.819 | 0.631 | 0.538 | 0.434 | 0.398 |

The curves on Plates IV, V, and VI show that less steam flowed through the divergent nozzles at the right than through the orifices at the left. Also in the case of the nozzle with rounded entrance the maximum rate of flow was reached by the time the ratio of back pressure to initial pressure reached the value 0.85. It seems from the curves on Figs. 52 to 56, and from data regarding orifices, that the pressure in general falls at the throat of the nozzle and then rises again. Experiments indicate that the pressure in the throat of the nozzle falls to that value which gives the maximum flow of steam by weight at any given initial pressure. By calculating the energy given up during the fall in pressure, the corresponding velocity may be ascertained, and the proper cross-sectional area for the smallest part of the nozzle may be found.

Referring to Fig. 26, to calculate the proper diameter of nozzle for the present example, where pres. = 112 pds. abs.,

The entropy $FL = 1.10$,
" " $FA = 1.06$.

Therefore the quality at $A = 0.96$ or 4% of the steam is condensed in passing the throat of the nozzle.

$$\begin{aligned} H_1 &= 844 + 24 &&= 868 \text{ B.T.U.} \\ H_2 &= E_{FA} \times T_2 = 1.06 \times 797 &&= 845 \quad \text{"} \\ H_1 - H_2 & &&= \phantom{0}23 \quad \text{"} \end{aligned}$$

Neglecting the loss that may have occurred up to the point under consideration,

Velocity in throat $= \sqrt{778 \times 64.4 \times 23} = 1070$ ft. per sec.
Specific volume at 112 pds. = 3.96 cu. ft.
Volume at quality 0.96 = 3.8 cu. ft.
Volume passing per second $= 0.25 \times 3.8 = 0.95$ cu. ft.

Area of cross-section $= \frac{0.95 \times 144}{1070} = 0.128$ sq. in.

Diameter required $=0.404$ inch or approximately 13/32″.

Having found the largest and smallest diameters of the nozzle the latter may be drawn to scale. The length must be decided upon according to circumstances and the designer's judgment as to the effect of length and angle of divergence upon the friction losses. The points in the length of the nozzle where the previously calculated pressures will occur may be located with the assistance of a pair of dividers for finding the diameters corresponding to the areas for their respective pressures.

Another form in which the problem may present itself is, given the initial and final conditions of the steam, to find what loss of energy will occur by reason of resistance in a given nozzle.

Let it be found from a test that at the end of expansion from 150 lbs. abs. to $1\frac{1}{2}$ lbs. abs. the quality of exhaust is 0.816. It is required to find the percentage of loss due to frictional resistance in the nozzle.

As before, $H_1=1090$ B.T.U. $H_2=800$ B.T.U.
$H_1-H_2=290$ B.T.U.

Quality at $A$ = quality due to adiabatic expansion $=0.774$.
Increase in quality represented by $AX=0.816-0.774=0.042$.
Hence, $y(H_1-H_2) \div H_v = 0.281y = 0.042$.

$$y = \text{loss of energy} = \frac{0.042}{0.281} = 0.15, \quad \text{or} \quad 15\%.$$

This problem being the inverse of the one previously worked out, the result just found is the same as the assumption of energy loss in the previous example.

The method developed in Chapter IV for simplifying computations of velocity by means of the heat diagram may be

used equally well in cases involving the allowance for losses. Thus, instead of equation (17),

$$V=\{64.4\times778(1-y)(H_1-H_2)\}^{\frac{1}{2}},$$

may be written

$$V=158\{(E_1+E_2)(T_1-T_2)(1-y)\}^{\frac{1}{2}}, \quad . \quad . \quad (19)$$

where $E_1$ and $E_2$ represent entropy changes at absolute temperatures $T_1$ and $T_2$ respectively, as before.

If the values of $y$ are known for a given type of nozzle operating under given pressures, the velocities may be predicted. It is necessary first, however, to analyze results obtained by experiment in order to find proper values for the coefficient $y$.

Suppose, for example, that curves representing actually obtained results from a given type of orifice or nozzle have been plotted. Curves $A$ on Plates II and III are of this character. Curves $B$ are plotted from equation (17), using the value $y=0$. The loss of velocity in the actual orifice or nozzle is then represented by the distance between the curves $A$ and $B$. Let it be required to find the friction loss $y$ at different initial pressures, and to use these values for obtaining a curve coinciding with curve $A$.

Let the velocity from the actual curve $A$ be called $V_a$;
" " " " " ideal " $B$ " " $V_b$.

Then
$$V_a=\sqrt{50103(H_1-H_2)(1-y)};$$

$$V_b=\sqrt{50103(H_1-H_2)};$$

$$\frac{V_a}{V_b}=\sqrt{1-y} \quad \text{or} \quad y=1-\left(\frac{V_a}{V_b}\right)^2.$$

Values of $y$ may be plotted, as is done at the bottom of Plates II and III, from calculations given at top of page 89.

These calculations apply to the curves $A$ and $C$, Plate III.

The curves show that as the initial pressure is decreased

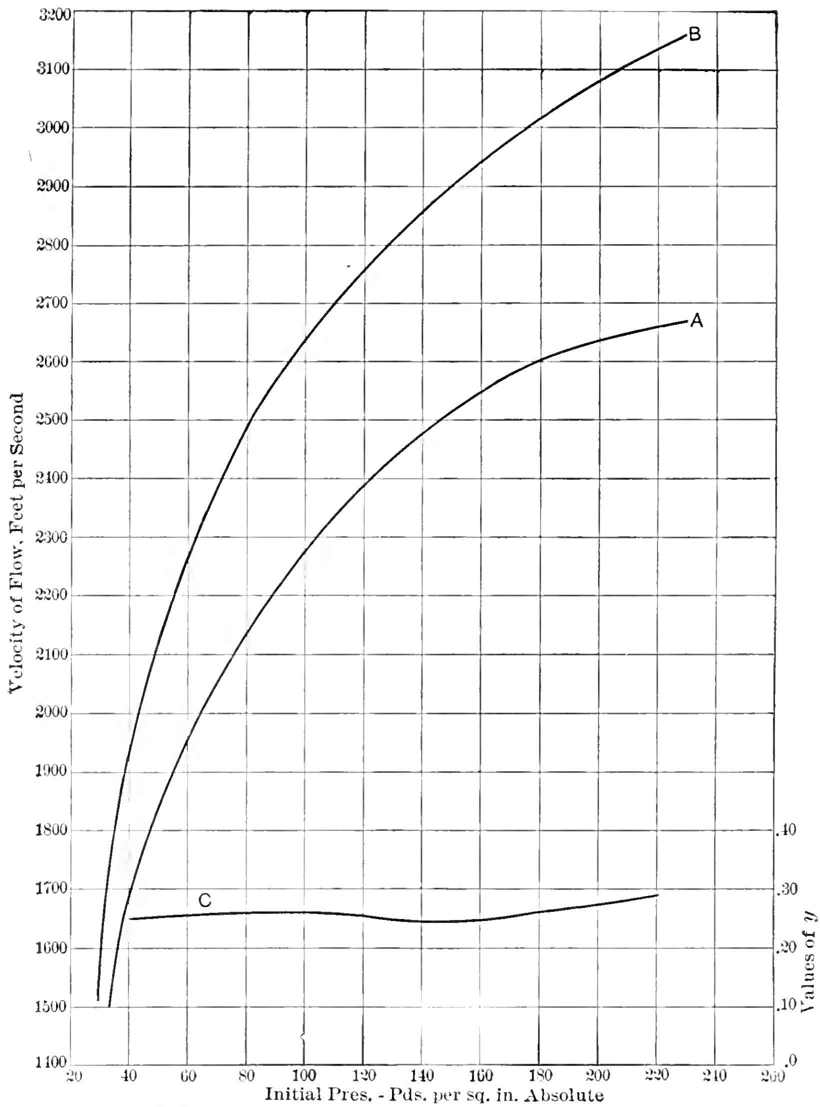
| Initia Pressure Absolute. | $V_a$. | $V_b$. | $\frac{V_a}{V_b}$. | $y=1-\left(\frac{V_a}{V_b}\right)^2$. |
|---|---|---|---|---|
| 60 | 1830 | 2260 | 0.81 | 0.44 |
| 100 | 2370 | 2640 | 0.90 | 0.19 |
| 140 | 2680 | 2870 | 0.94 | 0.12 |
| 180 | 2890 | 3030 | 0.95 | 0.09 |
| 220 | 3050 | 3140 | 0.97 | 0.06 |

the friction loss in the expanding nozzle increases, this being especially true for pressures below 100 pounds per square inch. In the case of the orifice in a thin plate, on the contrary, the losses are less at low pressures than at high pressures. The curve of losses on Plate II shows the value of $y$ to increase slightly with the pressure, but the change indicated is so small that the value of $y$ for this orifice may be regarded as constant at about 0.26. The values for the orifice and for the expanding nozzle are equal at about 80 pounds absolute initial pressure.

For the nozzles experimented with by Messrs. Jones and Rathbone the losses at 100 pounds and 50 pounds initial pressure absolute were as shown in the following table. The back pressure was atmospheric in all cases. In all the straight-bore nozzles the losses are higher for 100 pounds initial pressure than for 50 pounds, but in the case of the expanding nozzle the reverse is true, the value of $y$ at 100 pounds being only 40 per cent of that at 50 pounds.

| Diameter Nozzle, Inches. | Kind of Nozzle. | Initial Pressure, Pounds Absolute. | Loss Due to Friction. | |
|---|---|---|---|---|
| | | | Per Cent of Ideal. | Value of $y$. |
| $\frac{3}{16}$ | Straight bore, sharp entrance | 100 | 11.3 | 0.222 |
| $\frac{3}{16}$ | " " " " | 50 | 7.6 | 0.163 |
| $\frac{1}{4}$ | " " " " | 100 | 13.2 | 0.255 |
| $\frac{1}{4}$ | " " " " | 50 | 10.5 | 0.208 |
| $\frac{1}{4}$ | " " rounded " | 100 | 10.6 | 0.226 |
| $\frac{1}{4}$ | " " " " | 50 | 8.6 | 0.164 |
| $\frac{1}{4}$ | Expand. " " " | 100 | 5.7 | 0.125 |
| $\frac{1}{4}$ | " " " " | 50 | 16.7 | 0.312 |
| $\frac{3}{8}$ | Straight " sharp " | 100 | 7.5 | 0.145 |
| $\frac{3}{8}$ | " " " " | 50 | 3.6 | 0.077 |

PLATE II.

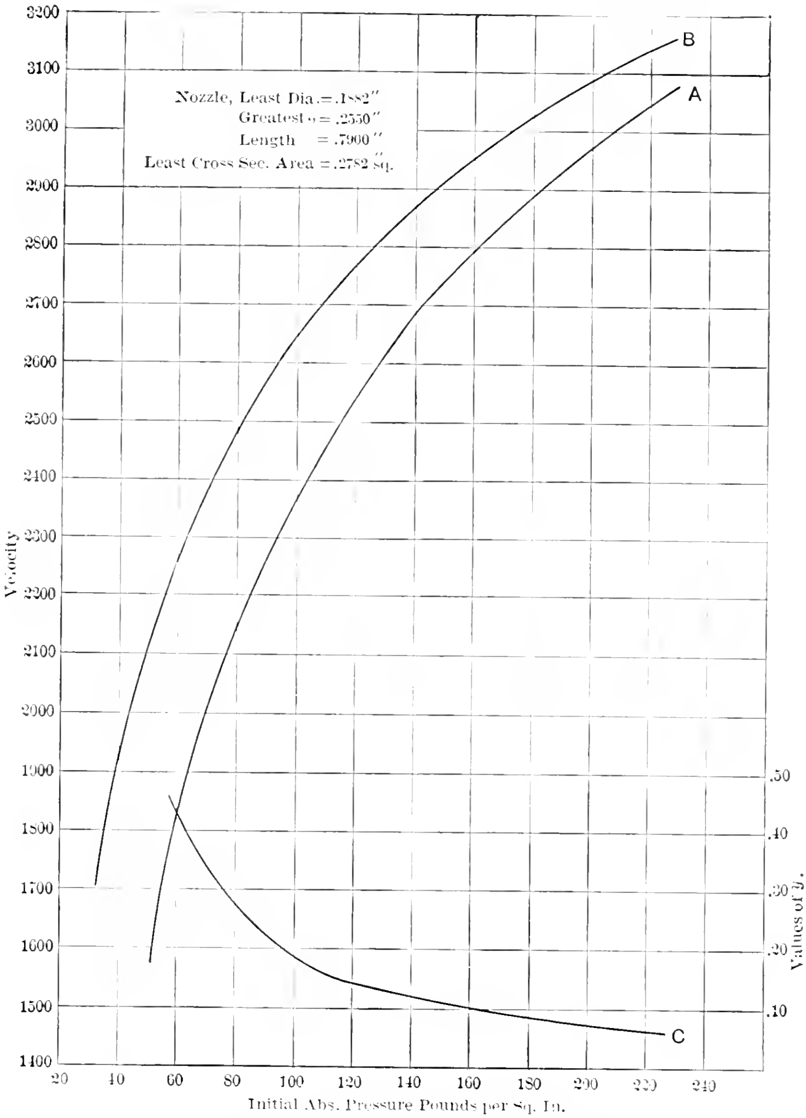



Curve *A*, Mr. Rosenhain's experiments; velocity corresponding to measured reaction of the jet from an orifice in a thin plate.

Curve *B*, calculated velocity, upon the assumption that all the heat energy concerned in the drop from the higher pressures before the orifice to the constant atmospheric pressure beyond the orifice was converted into the kinetic energy of the jet of steam.

Curve *C*, values of *y* at different pressures.

PLATE III.




Curve *A*, Mr. Rosenhain's experiments; velocity corresponding to measured reaction of the jet from an expanding nozzle. (Nozzle No. III A, p. 108.)

Curve *B*, calculated velocity, assuming that all the heat energy concerned in the drop from the higher pressures before the nozzle to the constant atmospheric pressure beyond was converted into the kinetic energy of the jet of steam.

CALCULATIONS FOR CURVES $A$ AND $B$ ON PLATE III.

Least diameter of nozzle.... 0.1882″ Length of nozzle......... 0.79″
Greatest diameter of nozzle. 0.2550″ Least area of cross-section. 0.2782″

| Initial Pressure, Pounds Absolute. | $T_1$ | $T_3$ | $T_1-T_3$ | $E_1+E_2$ | Pounds Discharg'd per Second as Measured. | Reaction Pounds, Observed. | Velocity from Reaction. | Velocity Ideal. |
|---|---|---|---|---|---|---|---|---|
| 35 | 720 | 673 | 47 | 2.67 | 0.013 | 0.45 | 1120 | 1770 |
| 55 | 748 | 673 | 75 | 2.55 | 0.021 | 1.10 | 1690 | 2180 |
| 75 | 768 | 673 | 95 | 2.48 | 0.028 | 1.80 | 2070 | 2420 |
| 95 | 785 | 673 | 112 | 2.42 | 0.038 | 2.70 | 2290 | 2600 |
| 115 | 799 | 673 | 126 | 2.37 | 0.044 | 3.45 | 2520 | 2730 |
| 135 | 811 | 673 | 138 | 2.34 | 0.052 | 4.25 | 2640 | 2840 |
| 155 | 822 | 673 | 149 | 2.31 | 0.059 | 5.05 | 2760 | 2930 |
| 175 | 831 | 673 | 158 | 2.28 | 0.066 | 5.85 | 2860 | 3000 |
| 195 | 840 | 673 | 167 | 2.25 | 0.073 | 6.65 | 2940 | 3060 |
| 215 | 849 | 673 | 176 | 2.23 | 0.079 | 7.45 | 3040 | 3130 |

# CHAPTER VI.

## EXPERIMENTAL WORK ON FLOW OF STEAM THROUGH ORIFICES, NOZZLES, AND TURBINE-BUCKETS.

IN the design of nozzles and steam-channels in general the following questions are involved:

(*a*) The weight of steam that will flow through when certain pressures exist at the inlet and outlet ends respectively.

(*b*) The velocity attained by the issuing jet of steam when a known weight per second is flowing.

(*c*) The heat expenditure necessary in order to produce a given amount of kinetic energy in the jet as it leaves the nozzle or passageway.

Experiments to determine the above have been made in various ways, and among the methods used are the following:

1. Steam caused to flow from a higher to a lower pressure through various shapes of orifice and nozzle, and the steam condensed and weighed. The results obtained by this method give the weight of steam that the orifices and nozzles will discharge per unit of time under differing inflow and outflow pressures This information, however, does not give the data for calculating the velocity attained by the steam, because the specific volume of the steam at different points along the nozzle depends upon the pressures at those points, and the latter are not known. Further, the nozzle allowing the greatest weight of steam to pass is not necessarily that giving the greatest velocity of outflow or the greatest energy of the jet.

2. Steam flowing as described in 1, but pressures along the nozzle investigated by means of a small "searching-tube" held axially in the nozzle. The tube has a small hole in its wall, and by moving the tube along the nozzle bore the hole occupies various positions and indicates on a gage connected to the end of the tube a more or less close approximation to the pressures existing at the points where the hole is brought to rest. It makes a considerable difference in the results, however, whether the hole in the tube is perpendicular to the axis of the tube or slants in the same direction as the flow of steam or in the opposite direction. Holes have also been drilled in the nozzle walls and pressures measured at those points. From such observations of pressures, the specific volume of the steam at various cross-sections has been calculated, and, the rate of steam-flow being known, the velocity at the different sections has been approximately found. This method is open to the objections that the accuracy of the pressure readings is very questionable, and the extent to which the steam fills out the cross-sectional areas of the nozzles is not known. However, much very valuable information has been obtained by this means as to the variation of pressure and the vibrations of the steam in the nozzle, the effect of varying back pressures, etc.

In experiments made in Sibley College during 1904–5 by Messrs. Weber and Law, the searching-tube was arranged so it communicated the pressure in the nozzle to the piston of a steam-engine indicator, and thus an autographic representation of the pressure changes was obtained. These experiments, and others along the same line, will be referred to later.

3. By arranging the nozzle so that as the steam flows out of it the reaction against the nozzle accompanying the acceleration of the steam can be measured, it is possible to ascertain the velocity the steam attains. The rate of steam-flow is measured by condensing and weighing, and the velocity in feet per second equals the reaction in pounds multiplied by $g$ $(=32.2)$ and divided by the weight of steam flowing per

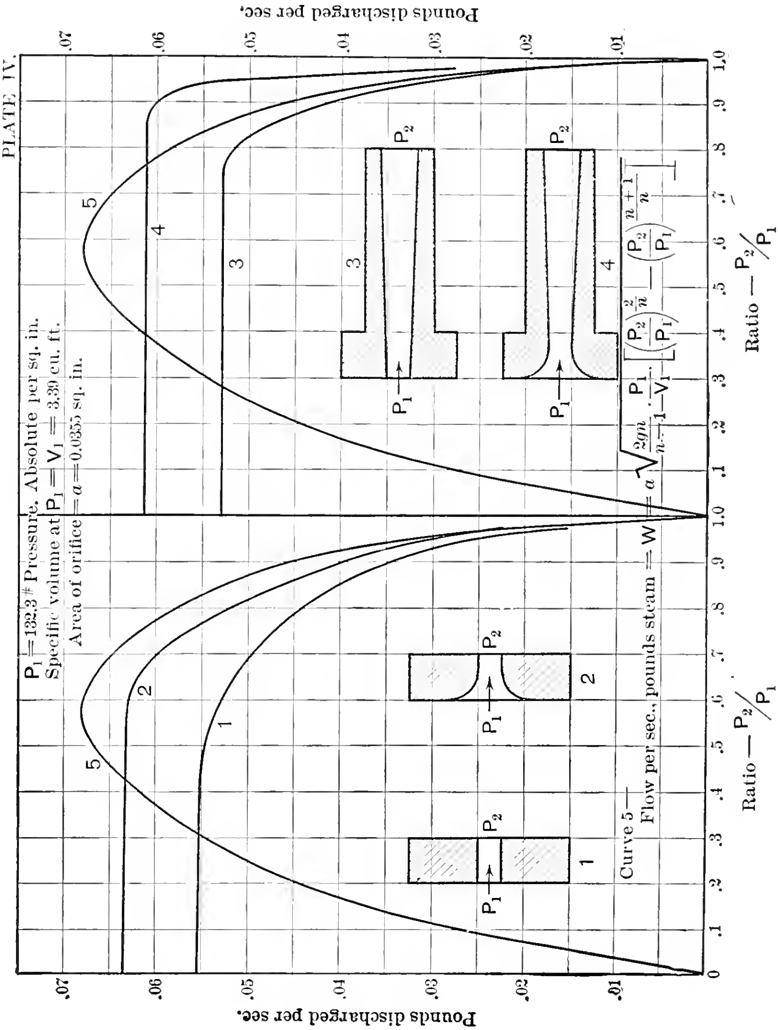
second. By measuring the weight and inlet and outlet temperatures of the condensing water, as well as the weight of condensed steam the heat given up in the nozzle can be found and the prime object of such experiments may be attained; that is, the efficiency of the nozzle may be found—or the amount of kinetic energy in foot-pounds that can be produced by one heat-unit in the entering steam.

4. If a nozzle delivering $W$ pounds of steam per second discharges into buckets having known entrance and exit angles, the velocity of the jet may be computed by means of formula 6 on page 12. See also plate facing page 128.

## Weight of Steam Flowing Through Orifices and Nozzles as Found Experimentally by Professor Gutermuth.

Curves 1, 2, 3, and 4, on Plates IV, V, and VI, show the weight of steam which flowed from the four orifices shown, for varying values of $\frac{p_2}{p_1}$ and for varying initial pressures. In each case more steam flowed through the orifice with the rounded entrance than through that with the sharp-edged entrance, and in each case the weight of steam flowing per second reached a maximum value, beyond which the weight per second did not increase or decrease as the pressure $p_2$ was decreased. The question of the flow of steam, by weight, depends upon the pressures immediately in the orifice, as well as upon those in the inflow and outflow vessels. Curves 5, which represent the adiabatic flow of a gas which has the same ratio of specific heats as dry and saturated steam, according to the equation developed in Chapter II, are not applicable to the case of steam-flow, unless the steam remains dry and saturated during expansion, or else is initially superheated and remains superheated during expansion. Steam in expanding adiabatically from a saturated condition becomes partially condensed,—the specific heat of the mixture changes and the flow is not like to that of a gas. If the steam remained superheated, or dry and saturated, during expansion, the

PLATE IV.


$P_1 = 132.3$ # Pressure, Absolute per sq. in.
Specific volume at $P_1 = V_1 = 3.39$ cu. ft.
Area of orifice $= a = 0.0355$ sq. in.
Curve 5—
Flow per sec., pounds steam $= W = a\sqrt{\frac{2gn}{n-1} \cdot \frac{P_1}{V_1} \cdot \left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$
Pounds discharged per sec.
Ratio — $P_2/P_1$
$P_1$
$P_2$
1
2
3
4
5
.01
.02
.03
.04
.05
.06
.07
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0

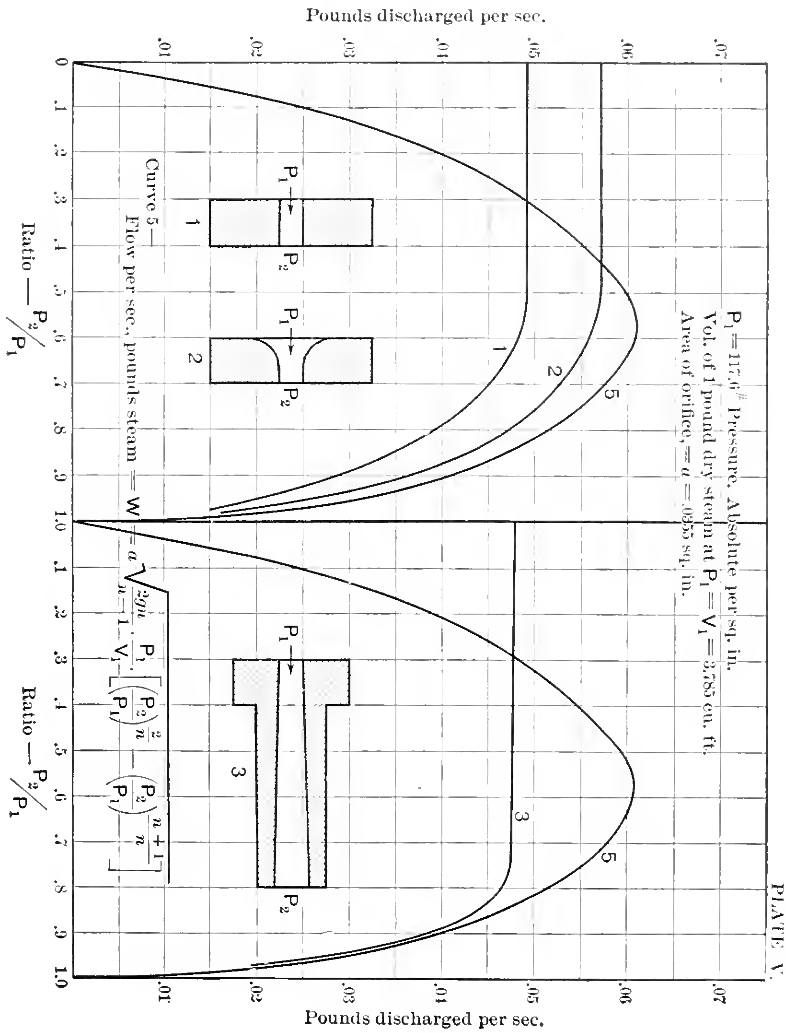
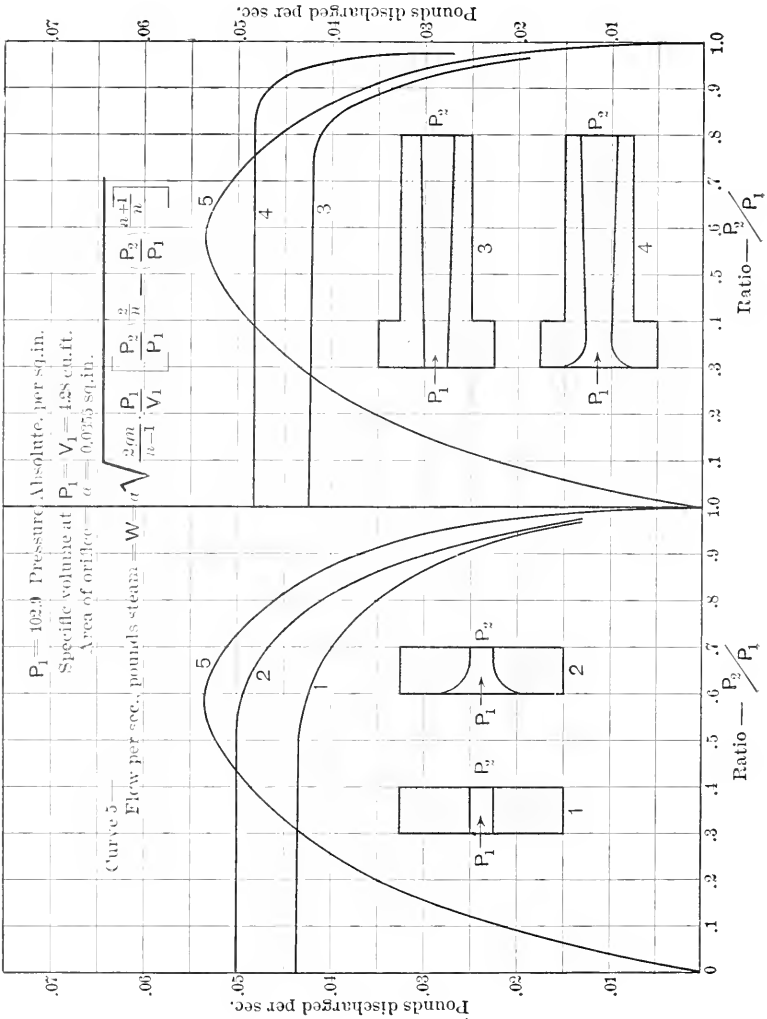


PLATE V.
Pounds discharged per sec.
$P_1 = 117.6$# Pressure. Absolute per sq. in.
Vol. of 1 pound dry steam at $P_1 = V_1 = 3.785$ cu. ft.
Area of orifice, $= a = .0355$ sq. in.
Curve 5—
Flow per sec., pounds steam $= W = a\sqrt{\frac{2gn}{n-1}\cdot\frac{P_1}{V_1}\cdot\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$
Ratio — $P_2/P_1$

PLATE VI.



$P_1 = 102.9$ Pressure Absolute per sq.in.
Specific volume at $P_1 = V_1$
Area of orifice $= a$
Curve 5—
Flow per sec., pounds steam $= W = a\sqrt{\frac{2gn}{n-1}\cdot\frac{P_1}{V_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$
Pounds discharged per sec.
Ratio — $P_2/P_1$

formula for the flow of gas would apply to that of the steam. As it is, however, the point at which the *maximum flow* of steam will occur, through an orifice having well-rounded entrance, agrees more or less closely with the indications of the equation for a gas, as is seen by the curves given, and with certain modifications the equation may be used to indicate the conditions of maximum flow. A very useful equation was developed by Mr. R. D. Napier, and modified by Professor Rankine, based upon experiments by Napier and the equation under discussion. The discharge through an orifice $a$ sq. ins. area from a pressure $p_1$ on one side to a lower pressure $p_2$ on the other side may be calculated as follows, according to Napier's formula:

$$W=\frac{ap_1}{70} \quad \text{if} \quad \frac{p_2}{p_1}= \text{or is less than } 0.60.$$

When $\dfrac{p_2}{p_1}$ is greater than 0.60, $W=\dfrac{ap_2}{42}\sqrt{\left\{\dfrac{3(p_1-p_2)}{2p_2}\right\}}$.

Thus, in the case of curve 2, Plate IV, the discharge according to this expression would be

$$W=\frac{0.0355\times 132}{70}=0.0669 \text{ pound per second.}$$

The observed maximum flow is 0.063+pounds, or about 94% of that given by the equation.

Similarly, on Plate V, curve 2,

$$W=\frac{0.0355\times 118}{70}=0.060 \text{ pound.}$$

The observed maximum flow is 0.05, or about 95% of that given by the equation.

On Plate VI, curve 2,

$$W=\frac{0.0355\times 103}{70}=0.0523 \text{ pound.}$$

The observed maximum flow is 0.050, or about 95.5% of that given by the equation.

The above equation may be taken as a guide for calculating the maximum flow of steam when the ratio $p_2 \div p_1$ is not greater than about 0.6, but it evidently does not apply closely unless the orifice has a well-rounded entrance.

It is to be observed that curves 4 on Plates IV and VI, for the divergent nozzles, show a smaller steam weight discharged per second than is discharged from the plain orifice 2. This, however, does not mean that the velocity in the divergent nozzle is less than that in the plain orifice.

The table opposite shows the results of experiments with the orifices on Plate VII, together with the calculations of the steam-flow by Napier's formula and by the thermodynamic formula which was developed in Chapter IV. All the experiments excepting those by Professor Peabody were made in the Sibley College laboratories under the direction of Professor R. C. Carpenter.

It has been shown in the preceding discussion that, at least for small diameters of opening, it is possible to calculate very closely the maximum weight of steam discharged per unit of time under given initial and final pressures. It has been quite thoroughly demonstrated that after a certain diminution of back pressure, the rate of flow, by weight, ceases to increase, and that it remains sensibly constant during further reduction of back pressure. The tables on page 109, calculated from the experiments of Mr. Walter Rosenhain, and of Mr. George Wilson, further confirm these statements.

The question as to the rate of increase of flow up to the maximum rate has been answered for convergent nozzles of certain sizes by the formula by Mr. R. D. Napier (see page 99), the work of Professor Rateau (see page 106), and that of Professor Gutermuth (see Plates IV, V, and VI).

The rate of flow, by weight, up to the point of maximum flow, depends very largely upon the shape of the inlet end of the orifice or nozzle,—whether the inlet is rounded or has

FLOW OF STEAM THROUGH ORIFICES.

| | Pressure above Atmosphere per Square Inch. | | | Barometer, Pounds per Square Inch. | Ratio of Absolute Pressures. | | Temperature of Steam in Second Chamber, ° F. | Quality of Entering Steam. | Flow in Pounds per Hour. | | | Apparent Coefficient of Flow. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | In First Chamber. | At Orifice. | In Second Chamber. | | Second Chamber to First Chamber. | Orifice to First Chamber. | | | By Experiment Weighed. | Cal'd by Thermodynamic Equation. | Calculated by Napier's Equation. | By Thermodynamic Equation. | By Napier's Equation. |
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 |
| | | | | | Orifice ($a_1$). Perry's Experiments. | | | | | | | | |
| 1 | 114.7 | 66.4 | −11.7 | 14.4 | .021 | .624 | 140.2 | 96.45 | 334.9 | 320.2 | 326 | 1.045 | 1.020 |
| 2 | 112.8 | 64.9 | +14.7 | 14.4 | .228 | .623 | 245.6 | 96.50 | 358.8 | 317.3 | 321.2 | 1.130 | 1.120 |
| 3 | 108 | 61.3 | 52.7 | 14.4 | .548 | .618 | 208.4 | 97.6 | 321.7 | 307.9 | 308.3 | 1.045 | 1.047 |
| 4 | 110.1 | 66.2 | 72.3 | 14.4 | .696 | .650 | 312.1 | 96.61 | 348.8 | 308.2 | 314.4 | 1.131 | 1.110 |
| 5 | 109 | 89.1 | 93.9 | 14.4 | .870 | .841 | 326.9 | 96 | 267.1 | 247.8 | 311.6 | 1.077 | .857 |
| 6 | 110.5 | 103.7 | 106.9 | 14.4 | .971 | .948 | 334.8 | 98.46 | 144.2 | 151.8 | 315.4 | .956 | .456 |
| 7 | 74.4 | 68.6 | 71.9 | 14.4 | .971 | .940 | 311.3 | 99.4 | 107 | 118.9 | 224.2 | .890 | .477 |
| 8 | 50.4 | 40.6 | 42 | 14.4 | .870 | .542 | 285.9 | 99.6 | 107 | 141.1 | 163.6 | .900 | .776 |
| | | | | | | Orifice ($a_2$). | | | | | | | |
| 1 | 111.9 | 59.2 | −10.9 | 14.39 | .037 | .583 | 148.6 | 97.26 | 330.9 | 315.9 | 318.9 | 1.047 | 1.037 |
| 2 | 114.8 | 61.2 | 14.8 | 14.39 | .226 | .585 | 248.4 | 96.7 | 336.5 | 319.7 | 326.5 | 1.052 | 1.03 |
| 3 | 99.3 | 57.2 | 52.4 | 14.43 | .587 | .634 | 205.7 | 96.9 | 289.1 | 283.8 | 287.2 | 1.02 | 1.0006 |
| 4 | 112.5 | 70 | 72.3 | 14.43 | .683 | .668 | 312.4 | 96.6 | 313.8 | 274.2 | 320.4 | 1.136 | .987 |
| 5 | 115.5 | 93.3 | 93.2 | 14.43 | .828 | .830 | 326.0 | 96.4 | 262 | 263.1 | 328.1 | .994 | .798 |
| 6 | 112.6 | 109.2 | 108.8 | 14.43 | .970 | .978 | 336.5 | 98.5 | 38.9 | 114.2 | 320.8 | .34 | .121 |
| 7 | 94.7 | 50.5 | 14.8 | 14.43 | .267 | .501 | 247.1 | 96.6 | 286.3 | 271.4 | 275.6 | 1.054 | 1.037 |
| 8 | 74.7 | 39.4 | 14.3 | 14.43 | .321 | .604 | 240.9 | 97.3 | 240.3 | 224.6 | 225 | 1.068 | 1.067 |
| 9 | 53.4 | 26 | 15 | 14.43 | .433 | .505 | 248.2 | 99.3 | 182.2 | 171.2 | 171.3 | 1.063 | 1.064 |
| | | | | | | Orifice ($a_3$). | | | | | | | |
| 1 | 116.1 | 63.8 | − 9.6 | 14.3 | .039 | .505 | 164.2 | 96.7 | 312.5 | 324.4 | 329.3 | 1.059 | 1.04 |
| 2 | 96 | 51 | −11.7 | 14.3 | .025 | .503 | 140 | 96.5 | 291.4 | 273 | 278.5 | 1.067 | 1.046 |
| 3 | 75.1 | 39 | −10.9 | 14.3 | .038 | .506 | 140.6 | 97.1 | 235 | 225.4 | 225.7 | 1.042 | 1.041 |
| 4 | 52.7 | 26 | −10.5 | 14.3 | .056 | .601 | 157.4 | 99.4 | 179.5 | 170.1 | 169.2 | 1.055 | 1.060 |
| 5 | 102.1 | 55.1 | 15 | 14.3 | .251 | .506 | 246 | 97.2 | 303.5 | 290.7 | 293.9 | 1.044 | 1.032 |
| 6 | 113 | 61.7 | 62.7 | 14.35 | .526 | .597 | 296.7 | 97.6 | 327 | 314 | 321.4 | 1.041 | 1.010 |
| 7 | 112.4 | 65.3 | 69.7 | 14.35 | .662 | .630 | 309.9 | 97.3 | 318.9 | 306.3 | 320.2 | 1.042 | .995 |
| 8 | 115.4 | 88.7 | 91.3 | 14.35 | .814 | .800 | 306.2 | 96.6 | 274.9 | 280.6 | 327.8 | .979 | .838 |
| | | | Results of Professor Peabody's Experiments. Orifice ($b_1$). | | | | | | | | | | |
| 1 | 74.1 | 41.2 | 14.8 | 14.7 | .332 | .630 | 259.2 | 98.8 | 221 | 217 | 224 | 1.018 | .986 |
| 2 | 71 | 39.6 | 13.2 | 14.8 | .336 | .634 | 281.7 | 98.5 | 213 | 207.8 | 215 | 1.025 | .99 |
| 3 | 72.6 | 40.6 | 19.7 | 14.7 | .394 | .634 | 284.5 | 99.5 | 216 | 211.4 | 220 | 1.022 | .982 |
| 4 | 75.9 | 42.6 | 20.4 | 14.7 | .387 | .632 | 283.4 | 99.7 | 228 | 219.3 | 227 | 1.04 | 1.004 |
| 5 | 71.9 | 40.6 | 24.5 | 14.7 | .454 | .638 | 285.1 | 99.7 | 213 | 209.7 | 218 | 1.016 | .977 |
| | | | | | | Orifice ($b_2$). | | | | | | | |
| 1 | 72.8 | 39 | 14.8 | 14.8 | .338 | .614 | 281.7 | 99.7 | 225 | 213.6 | 221 | 1.053 | 1.018 |
| 2 | 72.1 | 38.8 | 20.4 | 14.8 | .405 | .617 | 288 | 99.5 | 223.5 | 211.7 | 219 | 1.056 | 1.02 |
| 3 | 72.6 | 39 | 24.7 | 14.8 | .452 | .616 | 291.2 | 99.5 | 223 | 213.1 | 220 | 1.046 | 1.013 |
| 4 | 73.1 | 39.2 | 29.9 | 14.8 | .509 | .615 | 293.4 | 99.5 | 225.5 | 213 | 222 | 1.054 | 1.015 |
| | | | | | | Orifice ($b_3$). | | | | | | | |
| 1 | 72.6 | 36.1 | 24.8 | 14.9 | .454 | .583 | 288.8 | 99.6 | 225 | 213.5 | 220 | 1.054 | 1.022 |
| 2 | 72.6 | 36.1 | 10.0 | 14.9 | .308 | .583 | 286.0 | 99.6 | 225 | 213.5 | 220 | 1.054 | 1.022 |
| 3 | 72.7 | 36.2 | 14.9 | 14.8 | .339 | .583 | 282.9 | 99.6 | 227 | 213 | 220 | 1.066 | 1.031 |
| 4 | 126.3 | 69 | 27.8 | 14.7 | .295 | .504 | 311 | 99.5 | 358.8 | 338.9 | 355 | 1.058 | 1.01 |
| 5 | 125 | 67.9 | 40.8 | 14.7 | .398 | .508 | 314.6 | 99.9 | 355 | 334.8 | 352 | 1.06 | 1.01 |
| | | | Results of Experiments by Mickle and Kuhn. Orifice ($c$). | | | | | | | | | | |
| 1 | 87.9 | 43.4 | 12 | 14.5 | .250 | .57 | ..... | 97.8 | 877.3 | ..... | 1042.2 | ..... | .841 |
| 2 | 85.3 | 46.8 | 32.1 | 14.5 | .468 | .617 | ..... | 97.8 | 708.1 | | | | |
| 3 | 92.0 | 48.5 | 37 | 14.5 | .171 | .588 | ..... | 97 | 1007.1 | ..... | 1093.1 | ..... | 1.003 |
| 4 | 78.9 | 36.6 | .74 | 14.5 | .161 | .6 | ..... | 98.4 | 867 | ..... | 950.6 | ..... | .912 |
| 5 | 36.5 | 25.3 | −2.27 | 14.5 | .174 | .560 | ..... | 98.5 | 708 | ..... | 722.6 | ..... | .979 |
| 6 | 36.2 | 13.4 | −5.68 | 14.5 | .174 | .55 | ..... | 99.7 | 476.3 | ..... | 516 | ..... | .923 |
| 7 | 16.1 | .1 | −8.89 | 14.5 | .183 | .48 | ..... | 100.0 | 205.3 | ..... | 311.4 | ..... | .648 |

FIG. 1

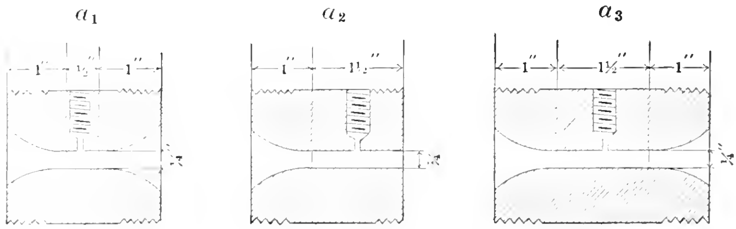



FIG. 2

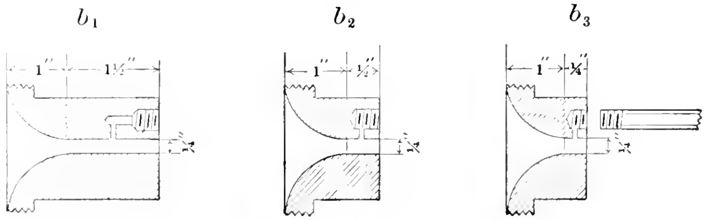
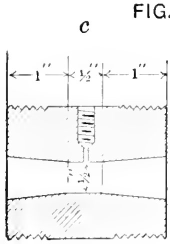



FIG. 3




ORIFICES FOR EXPERIMENTS ON P. 101.

PLATE VII.

square or sharp corners. The maximum rate of flow is reached much more quickly in some cases than in others, as is shown by Plates IV, V, and VI.

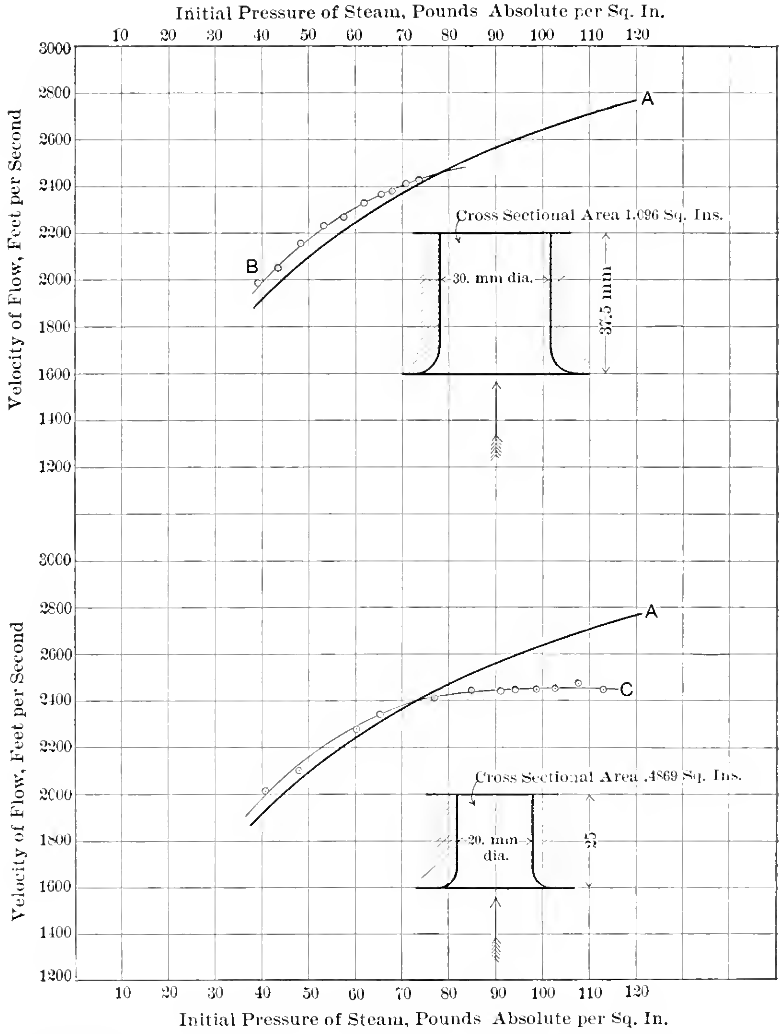
Orifices and nozzles having well-rounded entrances will pass more steam than those with sharp-cornered entrances, but this does not mean that they will emit a stream or jet having a correspondingly greater velocity than the latter. It seems that the rounded or bell-shaped inlet may cause a larger amount of steam to be admitted than can be efficiently expanded in the nozzle, and that a nozzle having its entrance only slightly rounded may have a higher efficiency than one with a large convergence of inlet.

In general, the shape of the inlet has greater influence upon the rate of discharge than has that of the outlet; while the outlet end has more influence upon the efficiency of expansion of the steam, and hence upon its exit velocity. The experimental work to be discussed later bears out these statements.

Whether or not the weight of steam flowing through orifices and passages of large size and more or less irregular shape can be calculated as satisfactorily as for the comparatively small sizes that have been used in experiments is not certain. The quantity of steam that will flow through a hole one square inch in cross-sectional area, for instance, is so great that experiments with such large orifices are seldom made. However, the experiments of Professor Rateau, and of Mr. George Wilson, given in the tables on pages 106 and 109, were made with openings from about $\frac{1}{2}$ inch diameter up to over an inch. Unless the source of steam-supply is of great capacity, experiments with openings of large area are of necessity made with comparatively low pressures.

Plate VIII gives velocities calculated from the reactions measured by Mr. George Wilson (London Engineering, 1872). The rate of flow was taken from the curve on Plate X. The inlet side of the orifices was made in the shape of what is called the "contracted vein," with the idea of passing the

PLATE VIII.




Curves *A* give velocity under ideal conditions of steam-flow into the atmosphere.

Curves *B* and *C* give calculated velocity as indicated by measured reaction. From experiments by Mr. George Wilson.

greatest possible volume of steam. The orifices were of comparatively large size (2 and 3 centimeters diam. respectively), and it may be that the weight of steam discharged per second was somewhat greater than that calculated and used in finding the velocity from the reaction. That would account, at least, for the calculated velocity being somewhat above that given by the ideal curves $A$, because the velocity is calculated from the equation

$$V=\frac{R\times 32.2}{W},$$

and therefore varies inversely as the weight of flow, $W$. However the curves show the same characteristics as the other results given for orifices and straight tubes, namely, a decided falling off in velocity for initial pressure above 70 or 80 pounds absolute, and comparatively high velocities for pressures lower than 70 or 80 pounds.

Further, comparing these curves with those from small nozzles for which the velocity has been determined by measurement of both reaction and weight of flow (see page 125), it seems safe to conclude that the velocities given on Plate VIII are not more than from 10 to 15 per cent too high, if indeed they are as much as that above the actual values. The surface of the orifice, causing frictional resistance to flow, increases only as the diameter of orifice, while the quantity of steam increases as the square of the diameter. It is therefore probable that with large orifices of favorable shape the frictional losses are proportionately less than with small orifices and nozzles, and that the high velocities indicated by the curves $B$ and $C$ were more closely realized than comparisons with results from smaller orifices and nozzles would lead one to believe.

The calculated results in the following table agree more closely with observed results in the case of the convergent nozzles than in that of the orifice in the thin plate. The convergent nozzles were simply orifices with bell-shaped entrances, and it was shown on page 99 that the equations for weight of

discharge apply more closely to such orifices than to those with sharp-cornered entrances.

RESULTS OF EXPERIMENTS BY PROFESSOR RATEAU, AND CALCULATIONS FROM THEM.

| Diameter, Inches. | Area, Square Inches, at 120° C. | $P_1$. Initial Pressure Absolute, Pounds per Square Inch. | $P_2 = 0.57 P_1$. | $V_2$. Volume after Adiabatic Expansion to $P_2$. | $T_1$. Absolute. | $T_2$. Absolute. | $T_1 - T_2$. | $E_1 + E_2$. | $W$. Weight of Steam, Pounds per Second, Actual. | Calculated Weight, Steam, Pounds per Second. | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 0.626 | 0.308 | 131.0 | 75.0 | 5.60 | 808 | 768 | 40 | 2.19 | 0.530 | 0.565 | A |
| " | " | 121.0 | 69.0 | 6.02 | 803 | 763 | 40 | 2.23 | 0.499 | 0.530 | |
| " | " | 120.0 | 68.5 | 6.10 | 802 | 762 | 40 | 2.23 | 0.490 | 0.525 | |
| " | " | 95.0 | 54.0 | 7.60 | 785 | 746 | 39 | 2.31 | 0.390 | 0.424 | |
| " | " | 64.0 | 36.5 | 11.0 | 758 | 723 | 35 | 2.44 | 0.270 | 0.285 | |
| 0.413 | 0.135 | 130.0 | 74.0 | 5.65 | 808 | 767 | 41 | 2.19 | 0.255 | 0.250 | A |
| " | " | 120.0 | 68.5 | 6.10 | 802 | 762 | 40 | 2.23 | 0.235 | 0.230 | |
| " | " | 110.0 | 63.0 | 6.56 | 795 | 756 | 39 | 2.27 | 0.218 | 0.210 | |
| " | " | 153.0 | 87.0 | 4.85 | 821 | 778 | 43 | 2.15 | 0.295 | 0.294 | |
| 0.792 | 0.495 | 59.0 | 33.6 | 4.42 | 752 | 718 | 34 | 2.48 | 0.361 | 0.424 | B |
| " | " | 54.5 | 31.0 | 11.80 | 747 | 713 | 34 | 2.49 | 0.319 | 0.392 | |
| " | " | 42.5 | 24.2 | 12.80 | 732 | 699 | 33 | 2.57 | 0.246 | 0.307 | |
| 0.953 | 0.720 | 34.7 | 19.8 | 16.30 | 719 | 688 | 31 | 2.65 | 0.370 | 0.367 | A |
| " | " | 28.4 | 16.2 | 19.70 | 708 | 678 | 30 | 2.70 | 0.308 | 0.303 | |
| " | " | 50.8 | 29.0 | 23.50 | 743 | 709 | 34 | 2.53 | 0.532 | 0.544 | |
| " | " | 57.4 | 33.0 | 12.10 | 750 | 716 | 34 | 2.48 | 0.591 | 0.603 | |

A, convergent nozzle. B, orifice in thin plate.

A large amount of data on the pressures existing at different points along steam-nozzles, and in jets from orifices, has been obtained by experiments, and such information has thrown a considerable amount of light on turbine operation. But given that sort of data alone, designers are almost as much at sea as before regarding the true efficiency of a nozzle or steam-passage and the actual velocity of steam-jets.

**The experimental work** giving the most direct and satisfactory evidence concerning the efficiency of steam-flow in nozzles and orifices has been that determining the reaction of the jet against the vessel from which it flows.

The work of Mr. George Wilson (see London Engineering,

Vol. XIII, 1872) and of Mr. Walter Rosenhain (see Proc. Inst. C. E., London, 1899) was of this character, and in both cases the experiments were evidently made with care. Mr. Wilson's apparatus is shown on p. 140. He did not measure the quantity of steam discharged, but did obtain a measure of the reaction accompanying discharge, under various initial pressures, into the atmosphere, with the various orifices which he employed. The second table on page 109 gives a few of Mr. Wilson's results for the purpose of comparing the observed reactions with those given by the use of the equation developed in the following pages.

Mr. Walter Rosenhain, of the University of Cambridge, has gone a step farther than did Mr. Wilson, as he has measured both the reaction and the rate of steam-flow. Mr. Rosenhain's experiments cover a wide range of initial pressures, but the final pressure is that of the atmosphere in all the experiments, as was the case with Mr. Wilson's experiments.

Experiments are at the present time being carried on in Sibley College, in which the reaction and weight of flow are measured, and in which the back pressure is carried down below the atmospheric pressure, as is the case in all condensing turbine plants. It is the purpose of the experiments to measure the heat in the discharge from nozzles in which known kinetic energy is developed, per pound of steam supplied, and thus to find the efficiency of the nozzles when discharging into the vacuum in the condenser.

Mr. Rosenhain's apparatus is shown in Figs. 27–29, and the first table on page 109 gives calculations based upon the flow from the simple orifice, No. 1.

These results are given to show the degree of approximation to be attained by the use of the equations for calculating the weight of flow and the reaction as explained in Chapter IV. The velocities as calculated are also given, and all the variables are further represented in the curves plotted on Figs. 30–40.

These experiments are of great importance in at least partially answering the questions stated on page 93. It is hoped that before long experimental results giving further information

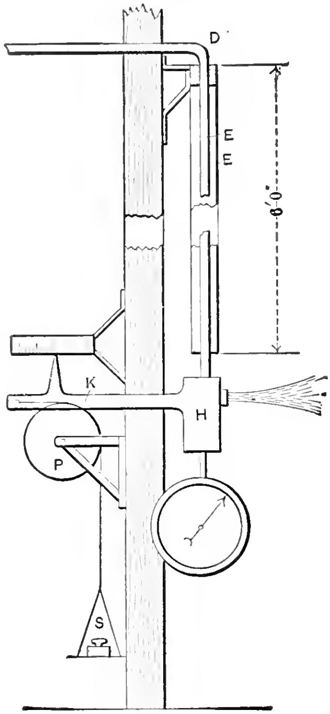



FIG. 27.

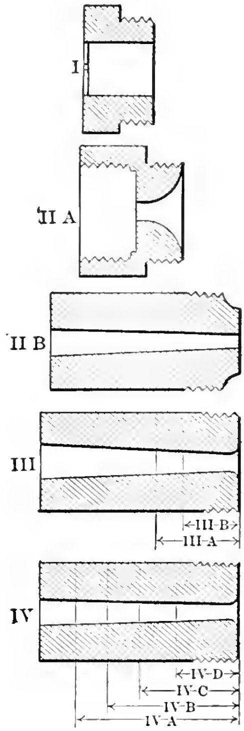



FIG. 28.

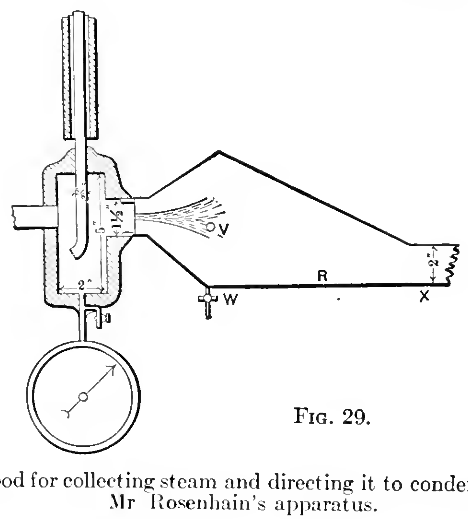



FIG. 29.

Hood for collecting steam and directing it to condenser.
Mr Rosenhain's apparatus.

will be available, especially regarding the flow into condensers maintaining conditions of vacuum.

EXPERIMENTS BY MR. WALTER ROSENHAIN.*

| $P_1$. Absolute Pressure, Initial Pounds per Square Inch. | Weight Flow per Second. | | Velocity of Efflux. | |
|---|---|---|---|---|
| | Calculated. | Actual. | Calculated from Calculated Reaction. | Calculated from Actual Reaction. |
| 26 | 0.0105 | | | |
| 35 | 0.0137 | 0.0137 | 1770 | 1580 |
| 55 | 0.0222 | 0.0220 | 2010 | 1900 |
| 75 | 0.0287 | 0.0290 | 2250 | 2100 |
| 95 | 0.0364 | 0.0370 | 2410 | 2250 |
| 115 | 0.0430 | 0.0440 | 2470 | 2350 |
| 135 | 0.0511 | 0.0510 | 2520 | 2450 |
| 155 | 0.0587 | 0.0580 | 2560 | 2530 |
| 175 | 0.0656 | 0.0640 | 2610 | 2580 |
| 195 | 0.0726 | 0.0720 | 2650 | 2620 |
| 215 | 0.0805 | 0.0780 | 2660 | 2650 |

Orifice in thin plate, with edges beveled towards direction of flow. Area of orifice 0.0275 sq. in., diam. 0.187 in. Calculated reaction $= R = (1.23P_1 - 14.7)$ pounds per square inch of orifice.

MR. GEORGE WILSON'S EXPERIMENTS.

| Diameter of Orifice, Inches. | Area of Orifice, Square Inches. | Absolute Initial Pressure. | Calculated Weight of Flow. | Calculated Reaction, Pounds. | Actual Reaction, Pounds. | Calculated Velocity. | |
|---|---|---|---|---|---|---|---|
| | | | | | | From Actual Reaction and Calculated Weight of Flow. | From Calculated Reaction and Weight of Flow, Ft. Sec. |
| 0.787 | 0.487 | 114.0 | 0.785 | 61.0 | 59.6 | 2450 | 2580 |
| " | " | 108.0 | 0.755 | 57.5 | 56.2 | 2400 | 2520 |
| " | " | 97.0 | 0.680 | 51.0 | 50.8 | 2400 | 2510 |
| " | " | 88.0 | 0.614 | 45.0 | 46.4 | 2440 | 2475 |
| " | " | 78.0 | 0.550 | 39.5 | 40.2 | 2360 | 2390 |
| 1.18 | 1.096 | 73.5 | 1.27 | 83.0 | 85.1 | 2160 | 2350 |
| " | " | 67.0 | 1.18 | 74.0 | 76.3 | 2080 | 2280 |
| " | " | 61.0 | 1.05 | 66.0 | 67.0 | 2050 | 2290 |
| " | " | 56.0 | 0.965 | 59.0 | 61.2 | 2050 | 2210 |
| " | " | 51.0 | 0.890 | 53.0 | 53.8 | 1950 | 2130 |

The calculated reaction given in the above tables was obtained by the use of the empirical formula developed on page 74, Chapter V, for jets discharging into the atmosphere. Thus,

$$\text{Reaction} = R = (1.23P_1 - 14.7) \text{ pounds per square inch of orifice.}$$

* Reviewed by permission of Mr. Rosenhain

Mr. Rosenhain starts with the premise justified both by theory and by experiment, that with a constant upper pressure a limiting velocity of efflux is reached when the lower pressure has been reduced to between 50 and 60 per cent of the higher pressure, while no limiting value is indicated when, with a constant low pressure, the higher pressure is increased. This does not apply to conically divergent nozzles, and the theoretical conclusions apply only to the narrowest section of a nozzle. The experimental conclusions apply only to orifices in thin plates or convergent nozzles of various types, including short cylindrical tubes.

Profiting by the records of previous experiments he decided that it would be desirable to measure the velocity of the steam as directly as possible, and to avoid estimating the density of the steam at the point of efflux. This estimation, depending upon temperature measurements, admits the greatest liability to error. Moreover, the velocity required for steam-turbine purposes is the actual velocity attained by the steam on leaving the nozzle, not merely a figure in feet per second from which the mass discharged could be calculated when the area of the orifice and the density of the steam are known. He found it necessary, therefore, to measure both the mass discharged and another quantity involving the velocity. For this second quantity he chose the momentum of the escaping jet. He first tried to measure this momentum by allowing the jet to impinge upon a semi-cylindrical bucket or vane in such a way as to reverse the jet, estimating that the pressure on the vane should then be equal to twice the momentum given to the jet per second. This method did not prove satisfactory and was rejected. He then adopted the reaction method.

Various methods of using the apparatus were tried, and, as a means of verifying the observations obtained by other methods, the method was adopted of obtaining the desired pressure at the gage by throttling the steam at the valve. The only observable difference he found between the jet at

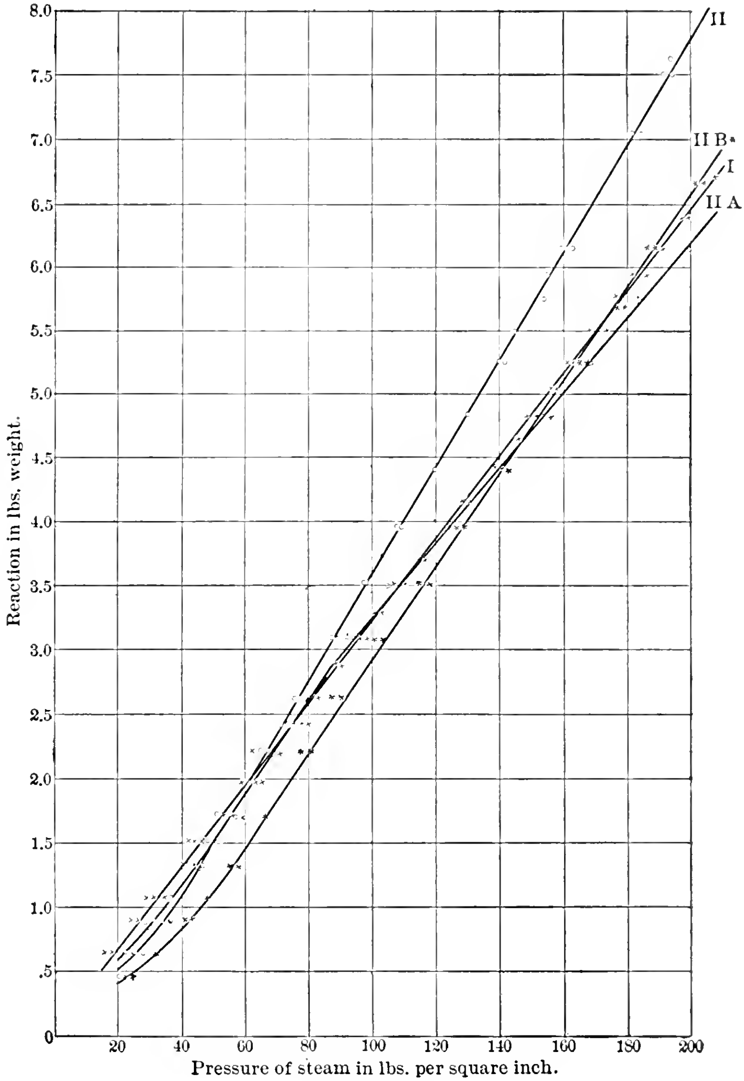

8.0
7.5
7.0
6.5
6.0
5.5
5.0
4.5
4.0
3.5
3.0
2.5
2.0
1.5
1.0
.5
0
20
40
60
80
100
120
140
160
180
200
II
II B
I
II A
Reaction in lbs. weight.
Pressure of steam in lbs. per square inch.

FIG. 30.

full way and by throttling to the same pressure was in the appearance of the jet. The throttled jet, when the throttling was considerable—as from 200 pds. per square inch to 20 lbs. per square inch—was of a darker color, much more transparent, but showing the brown color by transmitted light much more strongly; at the same pressure not the slightest difference in reaction could be observed between a "full-way" and a "throttled" jet.

The nozzles shown in section in Fig. 28 were of gun-metal, and were carefully prepared to exact dimensions. No. I is an orifice in a thin plate, produced by a very oblique chamfer on the outside. No. II consists of two parts drilled and turned up together. All the experiments with this nozzle as a whole were completed before the parts were separated to form the new nozzles IIA and IIB. Nos. III and IV were made of approximately the same length as IIB, and with larger and smaller tapers respectively. No. III was then cut down to form IIIA, the greatest diameter of which is equal to that of IV. Finally, IIIA was also cut down to form IIIB. No. IV was also cut down by $\frac{3}{8}$ inch at a time to form IVA, IVB, IVC, and IVD successively. In III and IV the inner edge of the nozzle is merely rounded off smoothly. These were designed on lines suggested by the results of the experiments on II, IIA, and IIB. The area of the orifice or nozzle does not enter into the calculation of the velocity. In order, however, to make the results strictly comparable, the entire set of nozzles was made with as nearly as possible the same least diameter, $\frac{3}{16}$ inch. This diameter and the tapers approximate to those used on a De Laval 5-H.P. turbine-motor. A table showing the dimensions of the nozzles as supplied with this turbine is given on page 114, for the sake of comparison. The actual least diameter of each nozzle was carefully measured with a micrometer microscope to an accuracy of 0.001 inch.

Fig. 31.

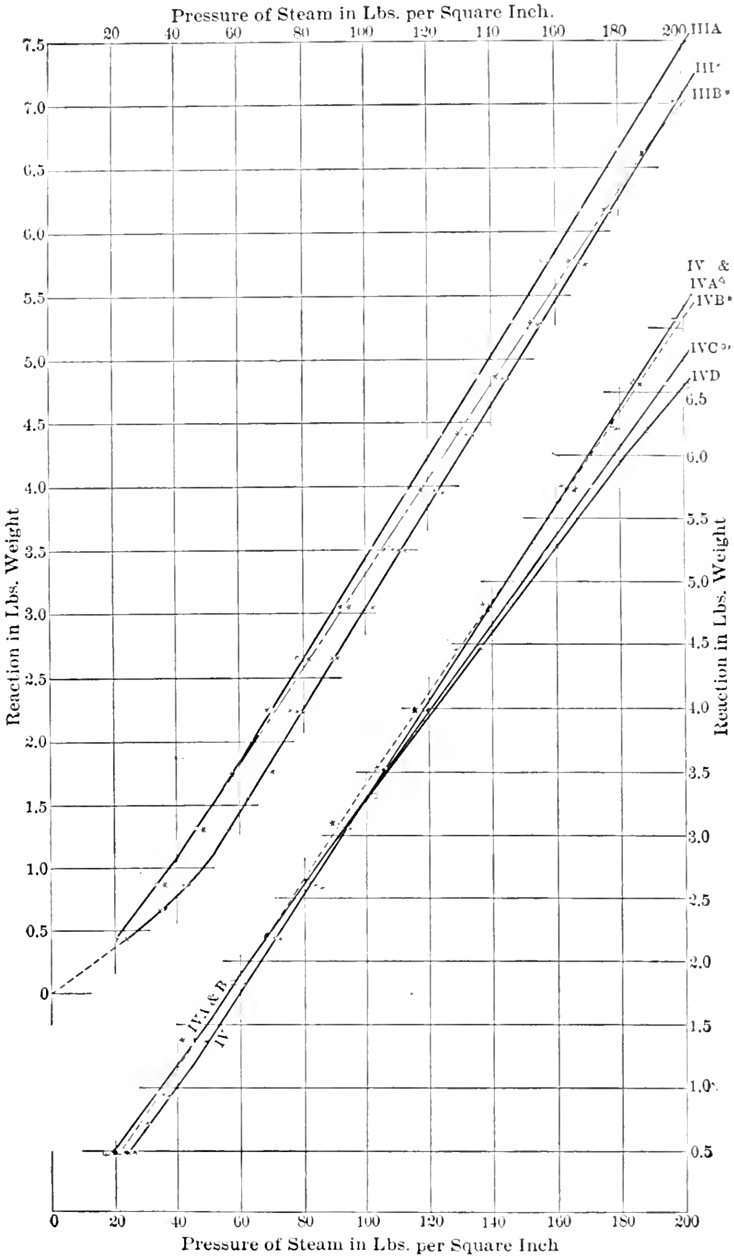

Pressure of Steam in Lbs. per Square Inch.
Reaction in Lbs. Weight
IIIA
III
IIIB
IV &
IVA
IVB
IVC
IVD
IVA & B
IV
Reaction in Lbs. Weight
Pressure of Steam in Lbs. per Square Inch

Fig. 32.

EXPERIMENTAL NOZZLES.

| Number. | Least Diameter. | Greatest Diameter. | Length. | Taper. | Remarks. |
|---|---|---|---|---|---|
| | Inches. | Inches. | Inches. | | |
| I | 0.1873 | ........ | ........ | ........ | Orifice in thin plate |
| II | 0.1840 | 0.287 | 2.1 | 1 in 20 | Compound nozzle |
| IIA | 0.1866 | ........ | 0.5 | ........ | Inlet half of II |
| IIB | 0.1849 | 0.287 | 1.6 | 1 in 20 | Outlet half of II |
| III | 0.1882 | 0.368 | 2.16 | 1 in 12 | Inlet edges slightly rounded |
| IIIA | 0.1882 | 0.255 | 0.79 | 1 in 12 | Inlet edges slightly rounded |
| IIIB | 0.1882 | 0.241 | 0.64 | 1 in 12 | Inlet edges slightly rounded |
| IV | 0.1830 | 0.255 | 2.16 | 1 in 30 | Inlet edges slightly rounded |
| IVA | 0.1830 | 0.242 | 1.785 | 1 in 30 | Inlet edges slightly rounded |
| IVB | 0.1830 | 0.230 | 1.41 | 1 in 30 | Inlet edges slightly rounded |
| IVC | 0.1830 | 0.217 | 1.035 | 1 in 30 | Inlet edges slightly rounded |
| IVD | 0.1830 | 0.205 | 0.66 | 1 in 30 | Inlet edges slightly rounded |

DE LAVAL NOZZLES FOR 5-HORSE-POWER TURBINE.

| Pressure. | Least Diameter. | Length. | Taper. |
|---|---|---|---|
| Pounds per Square Inch. | Inch. | Inch. | |
| 136 | 0.157 | 1.57 | 1 in 17.4 |
| 105 | 0.163 | 1.57 | 1 in 21.4 |
| Experiment IIB | 0.184 | 2.11 | 1 in 20.0 |
| 100 | 0.197 | 1.57 | 1 in 19.0 |
| 60 | 0.230 | 1.57 | 1 in 29.0 |
| 58 | 0.256 | 1.57 | 1 in 26.6 |

The formula used for the calculation of the velocity of the steam in the jet is

$$V=\frac{Rg}{W},$$

where $V$ is the velocity of the steam in feet per second;

$R$ is the reaction in lbs. weight;

$g$ is the acceleration of gravity taken at 32.2 feet per second per second;

$W$ is the weight of steam discharged in lbs.

From the description of the experiments it will be seen that $R$ and $W$ are measured directly. For purposes of calculation, points were plotted on squared paper showing for each nozzle

(*a*) Steam pressure as abscissa, $R$ as ordinate;

(*b*) Steam pressure as abscissa, $W$ as ordinate.

From the smooth curves drawn to represent these points values of $W$ and $R$ were taken and used in the above formula

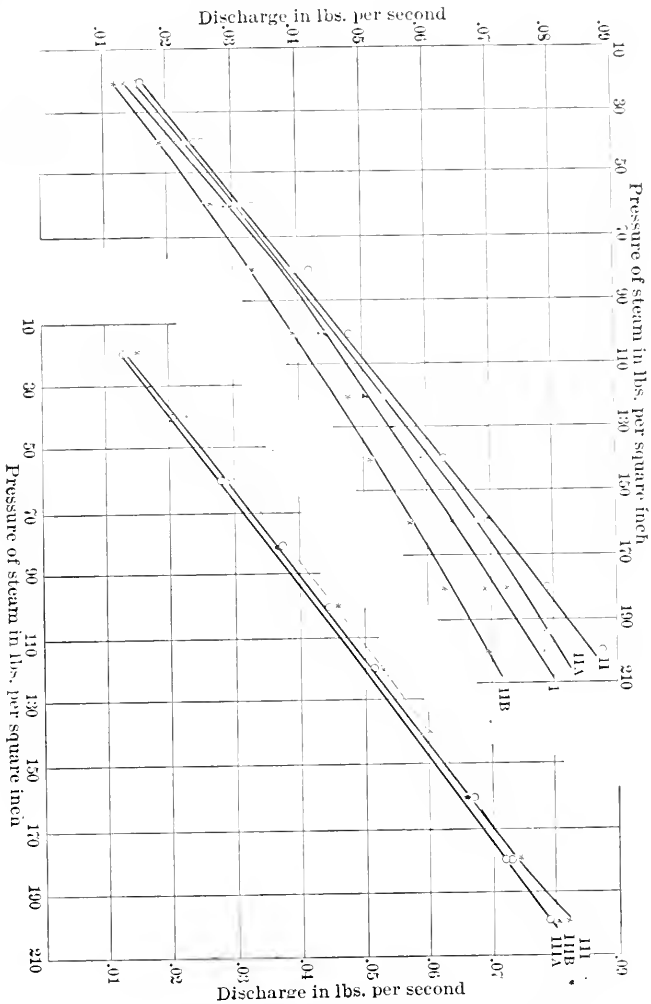



FIG. 33.

FIG. 34.

to give values of $V$; and, finally, a third curve was plotted, showing

(*c*) Steam pressure as abscissa, $V$ as ordinate.

This last curve represents the relation between pressure and

velocity, and also serves as a check on the accuracy of the arithmetical calculations.

The formula used assumes that at the point where the velocity is measured the steam has reached atmospheric pressure, otherwise the reaction would be increased by the remaining pressure; that is, the velocity here determined is that which the steam attains on reaching atmospheric pressure where this occurs outside the nozzle, or its velocity on leaving

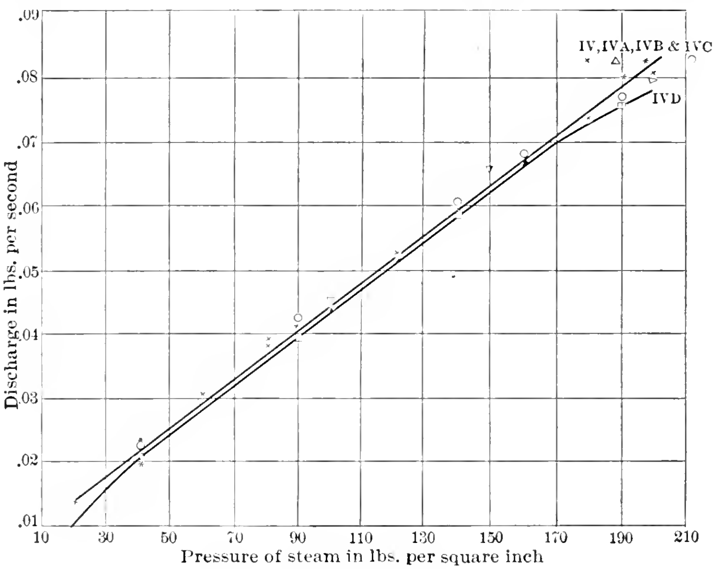



FIG. 35.

the nozzle where atmospheric pressure has been attained within the nozzle, in which case friction against the nozzle after complete expansion has occurred may cause the steam to lose some of its momentum. For practical purposes Mr. Rosenhain assumes that the velocities here found correspond to the kinetic energy of the jet on leaving the nozzle, an assumption which he found justified by observations on the shape of the jets. With the exception of those from the two very short nozzles, No. IIIB and No. IVD, the jets,—even that from No. I,—are very nearly parallel for several inches from

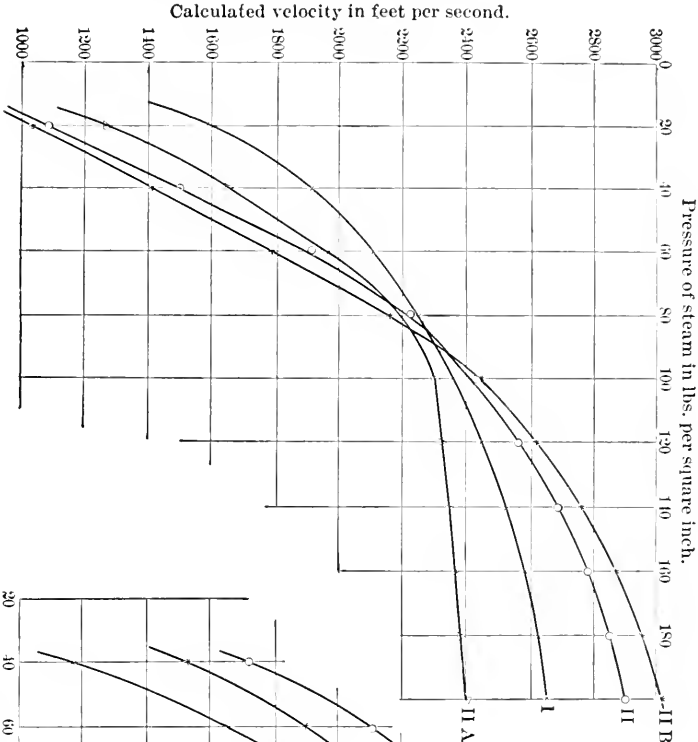

Calculated velocity in feet per second.
Pressure of steam in lbs. per square inch.
II B
II
I
II A

Fig. 36.

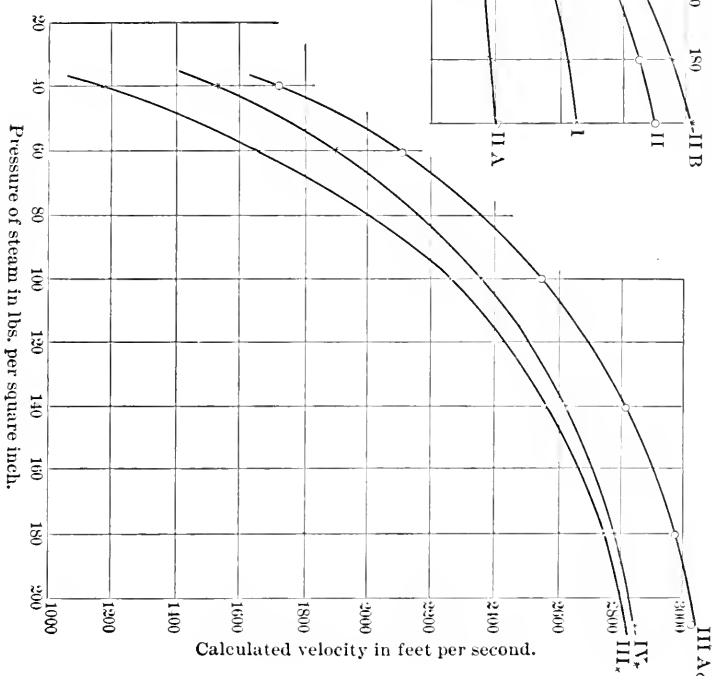

Pressure of steam in lbs. per square inch.
Calculated velocity in feet per second.
III A
IV
III

Fig. 37.

the end of the nozzle, or at most diverge at approximately the same taper as the nozzle.

In the case of the expanding nozzles this shows that the steam is expanded to atmospheric pressure before leaving the apparatus.

The first series of curves, Figs. 30, 33, and 36, represent the experiments made with nozzles Nos. I, II, IIA, and IIB. The reaction curves, Fig. 30, are mostly straight lines, i.e., the reaction is simply proportional to the pressure, but the constants vary for different nozzles. In the case of No. I, the orifice in a thin plate, the curve is a straight line through the origin, while for all other nozzles the line could reach the origin only through a curve. With IIA there is a slight but distinct sinuosity in this curve, and the points of IIB show a tendency to something similar. Mr. Rosenhain verified this by repeating the experiments under different conditions. He assigns the cause of the peculiarity to friction, as the sinuosity occurs only in those two nozzles where the friction would be large. It should be remembered, in comparing the curves, that the minimum diameters of II, IIA, and IIB are identical, but that of I differs very slightly.

The discharge curves (Fig. 33) occupy natural positions. The nozzle having an easy inlet and an expanding outlet gives the greatest discharge, the inlet being evidently more important than the outlet, hence the near approach of IIA to I.

The position of IIB so far below I would seem to justify Mr. Rosenhain's conclusion that "the sharp inlet is unsuited to passing a large quantity of steam through an expanding nozzle; while, on the other hand, the velocity curves (Fig. 36) show that the quantity of steam passed by a nozzle depends very considerably on the shape of the inlet, and the velocity of the steam on leaving the nozzle depends more on the shape of the outlet portion."

From this he concludes that the density of the steam at the narrowest section depends upon the shape of the inlet, and that "this density for a given internal pressure is greater

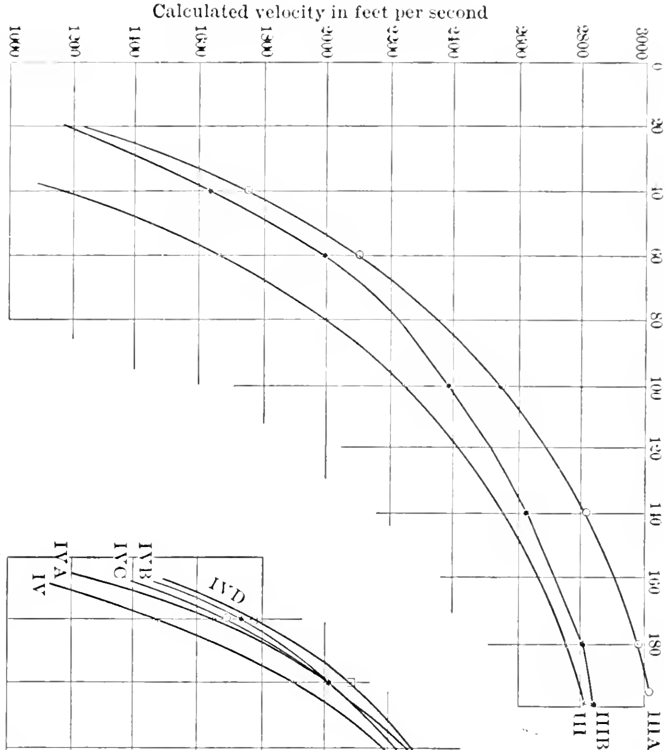

Calculated velocity in feet per second
Pressure of steam in lbs. per square inch
IIIA
IIIB
III

Fig. 38.

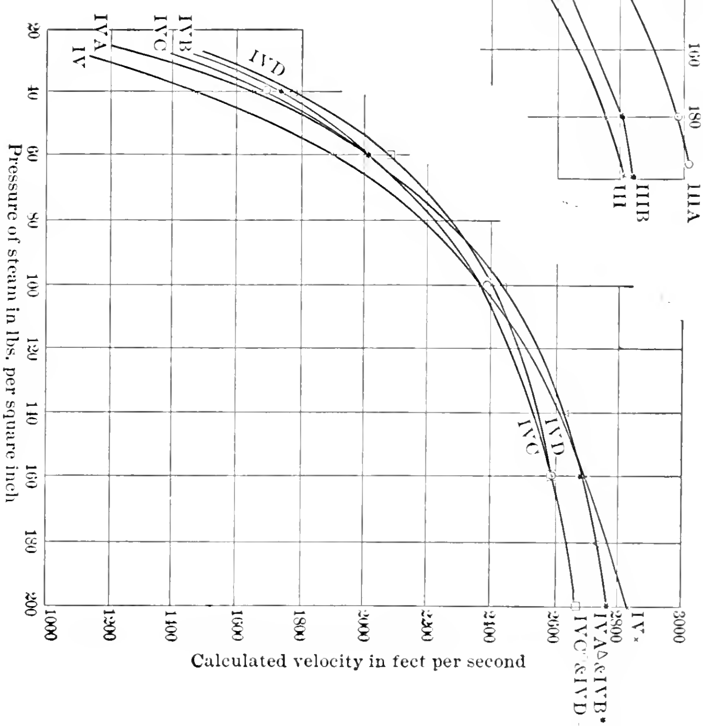

Calculated velocity in feet per second
Pressure of steam in lbs. per square inch
IVD
IVB
IVC
IVA
IV
IV×
IVA△&IVB•
IVC○&IVD□

Fig. 39.

with a well-rounded inlet than with a nozzle having a sharp inner edge."

This would account at once for the most conspicuous feature of this set of velocity curves, viz., that up to a pressure of about 80 lbs. per square inch the greatest velocity is attained by a jet from an orifice with a thin plate; above 100 lbs. per sq. inch, IIB, having a sharp inlet, gives a greater velocity than II, which has a rounded inlet and the same outlet. So that apparently a rounded inlet admits a greater weight of steam to the narrowest section than the nozzle can deal with efficiently. Thus, the advantage of I over IIA arises from its smaller discharge, which can expand with greater freedom and so develop a greater velocity than the denser steam issuing from IIA.

Considering the kinetic energy developed per pound of steam, the velocity curves may be taken to represent the "efficiency" of the various nozzles. From that point of view, Mr. Rosenhain concludes: "The effect of a sharp inlet is to reduce the density of the steam at the narrowest section, and hence less steam is passed, but the steam that does pass is fully or almost fully expanded; hence, though the discharge is reduced, the efficiency is increased."

In consequence of this conclusion, he designed all the later nozzles with an inner edge only slightly rounded off.

Nozzle IV was cut down by small steps, $\frac{3}{8}''$ being taken off the length each time, thus producing nozzles IVA, IVB, IVC, and IVD. Figs. 32, 35, and 39 show the reaction, discharge, and velocity at the nozzles. In order to present the results more clearly the curves of Fig. 40 were plotted. Here the length of nozzle is taken as abscissa, and reaction, discharge, and velocity are taken as ordinates for separate curves which have been plotted for steam pressures of 50, 100, 150, and 200 pounds (by gage) pressure respectively. "These curves show that reaction and discharge are influenced by the length of the nozzle in opposite ways. Very long nozzles with low steam pressure, or, more generally, nozzles that tend

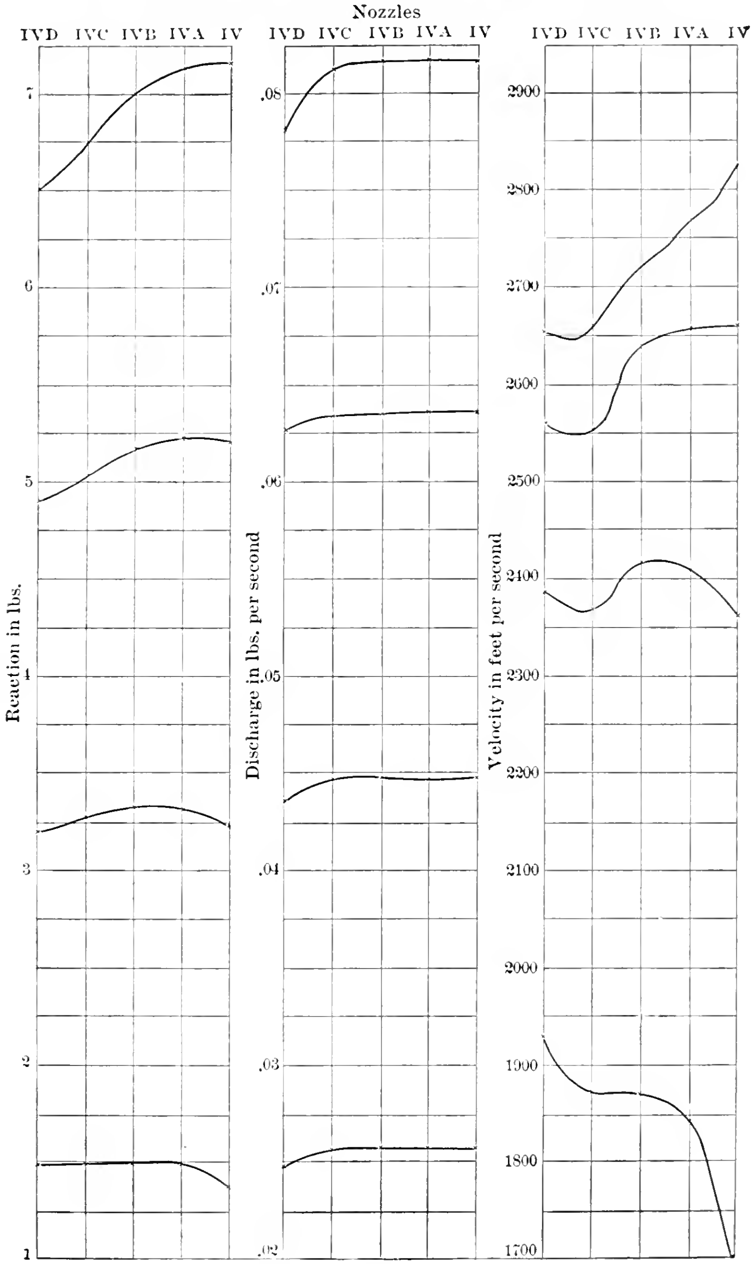

Nozzles
IVD IVC IVB IVA IV
IVD IVC IVB IVA IV
IVD IVC IVB IVA IV
Reaction in lbs.
7
6
5
4
3
2
1
Discharge in lbs. per second
.08
.07
.06
.05
.04
.03
.02
Velocity in feet per second
2900
2800
2700
2600
2500
2400
2300
2200
2100
2000
1900
1800
1700

FIG. 40.

to cause over-expansion, produce a large discharge but comparatively small reaction."

Considering further the question of "efficiency" in the sense just defined, it will be seen that the most efficient form of nozzle varies with the pressure. The reaction curve at 100 lbs. per square inch shows a maximum at IVA which recurs much more markedly in the corresponding velocity curve. The shape of the curve at 50 lbs. per square inch indicates that for these low pressures a long expanding cone is distinctly bad; in fact, a comparison of Figs. 36, 37, 38, and 39 shows that up to 80 lbs. per square inch an orifice in a thin plate is more efficient than any form of nozzle used in these experiments.

At 100 lbs. per square inch the velocity curve shows both a maximum and a minimum. A maximum was to be expected; the minimum would seem to indicate that the increase of length from IVD to IVC brings the discharge up to the highest value attainable for this pressure, while neither IVC nor IVB is long enough to develop the full reaction. Again, the fall in the velocity curve from IVA to IV he attributes to "over-expansion," especially as it disappears at 150 lbs. per sq. inch. Here the minimum has moved towards IVD, and it practically disappears at 200 lbs. per square inch. At 150 lbs. per square inch IV seems just to touch the maximum velocity attainable by a nozzle of that taper, while for 200 lbs. per sq. inch, even IV may be said to give insufficient expansion.

As a guide to the design of the most efficient nozzle, then—that is, the one that will develop the greatest kinetic energy in the jet per pound of steam consumed—Mr. Rosenhain summarizes the results of the experiments as follows:

"Up to a boiler pressure of about 80 lbs. per square inch, and for discharge into atmospheric pressure, the most efficient form is an orifice in a thin plate. For higher boiler pressures an expanding conical nozzle with an inner edge only slightly rounded should be used. The taper should not be very different

from 1 in 12, and the proper ratio of greatest and least diameters is given, according to present results, in the following table:

| Steam pressure, lbs. per sq. inch, gage | 80 | 100 | 140 | 160–200 |
|---|---|---|---|---|
| Ratio of diameters | 1.26 | 1.26–1.33 | 1.36 | 1.36 |

"The bearing of the above results on the thermohydrodynamic equation of Weisbach is not very direct. The part played by friction in these nozzles is very great and can only be allowed for in the equations by the introduction of artificial coefficients, and these hardly seem worth calculating, especially as it seems doubtful if hydrodynamic equations are applicable to gases. Hydrodynamics is based on the assumption of a perfectly homogeneous fluid, but a gas, and still less a vapor carrying particles of water in suspension, does not satisfy this condition."

## EXPERIMENTS WITH TURBINE-BUCKETS.

Extensive experimental turbine work was done in the Sibley College Laboratories during the years 1897–98–99, under the direction of Mr. Thomas Hall of the class of 1894, one of the designers of the Hall and Treat quadruple expansion engine. Mr. Hall held the Sibley Fellowship during 1894–95, and was subsequently an instructor for two years. During this latter period he superintended the experimental work discussed in the following pages, and to his efforts, supplemented by the efficient work of Messrs. Rathbone and Jones, '97–'98, and Messrs. Loetscher and McDonald, '98–'99, is to be given full credit for the valuable information obtained. The curves presented here have been plotted from the data obtained, some of the curves being given as originally plotted by the investigators.

The points investigated were as follows:

(*a*) The weight of flow of steam through nozzles of varying size under different initial steam pressures and atmospheric exhaust pressure.

(*b*) The actual velocity of the jet from the nozzles as indicated by the nozzle reaction.

(*c*) The impulse exerted by the jet upon buckets having various angles of entrance and exit.

(*d*) The impulse as affected by bucket-spacing.

(*e*) The impulse as affected by clearance between the nozzle and the buckets.

(*f*) The impulse as affected by placing a varying number of rows of stationary buckets in front of a set of movable buckets.

(*g*) The impulse as affected by the clearance between rows of buckets

(*h*) The substitution of air for steam, comparing the impulsive pressures upon the buckets in the two cases.

(*k*) The impulse as affected by "cutting over" the edges of the buckets by the jet of air from the nozzle.

(*l*) The efficiency of rough surface buckets as compared with those having smooth surfaces.

**The discharge from nozzles and buckets** was in all cases at atmospheric pressure. The nozzles experimented with were of diameters $\frac{1}{8}''$, $\frac{3}{16}''$, $\frac{1}{4}''$, and $\frac{3}{8}''$, 2 inches long, with rounded entrance and with sharp entrance, and with straight and expanding bores. The curves are marked so as to show to what character of nozzle they correspond. The weight of flow per second corresponds with the data previously given, and is given with other data for $\frac{1}{4}''$ nozzles in Fig. 41. The curves for the $\frac{1}{4}''$ nozzles show that for initial pressures up to about 70 pounds absolute the straight nozzles gave higher velocities than the expanding nozzle, but that above 70 pounds the reverse was true. However, in these cases the jet from the straight nozzles acted upon the buckets more efficiently than did that from the expanding nozzle.

The centers of the ends of the straight and the expanding nozzles were placed at the same distance from the buckets, and since the jet begins to diverge in the bore of the expanding nozzle, and not until it has left the straight nozzle, the experimenters concluded that the expanding nozzles hould be

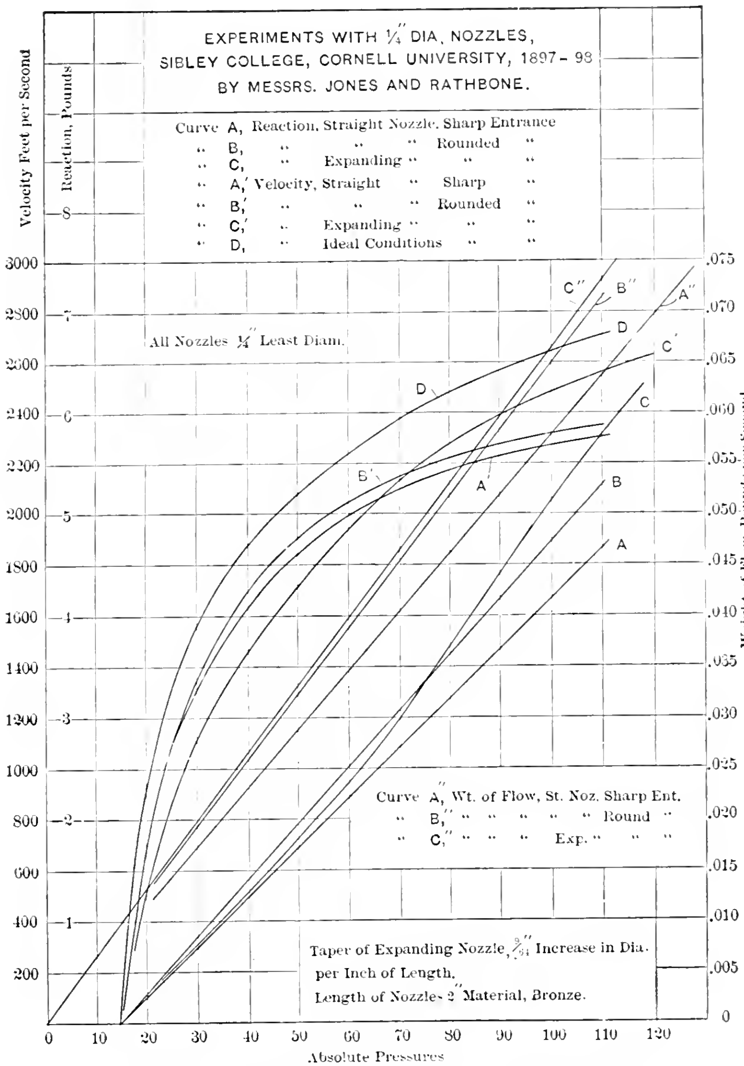



FIG. 41.—Note the difference in character of curves $C$ and $C'$, which are from the expanding nozzle, from the curves representing the straight-nozzle results. The energy of the jet from the expanding nozzle is below that from the straight nozzle up to about 70 pounds absolute, after which it goes above.

placed nearer the buckets than the straight nozzle for equal efficiency.

In general, the impulse upon the 135° buckets, Figs. 42 and 43, was somewhat higher than that upon the 150° buckets. This may have been due to the fact that the latter were somewhat thicker than the former, and hence had less space between them for passage of steam. Upon the basis of the tests made and shown by the curves, it was decided to use 135° buckets in all the tests, and to place the nozzles at such an angle that the stream would enter tangentially to the bucket surfaces. Sufficient buckets were used in all cases so that all the stream from the nozzle impinged upon buckets. There were from four to six buckets used in each set.

The general arrangement of this apparatus used is given in Fig. 52.

The clamps for holding the buckets were guided and attached to the balance scales, so that the impulse might be measured. The reaction upon the nozzles was obtained in a similar manner for each steam pressure employed, and the rate of flow at each pressure was determined by a separate test in which the steam from the nozzle was led to a condenser and then weighed. Preliminary runs were made until the apparatus was in satisfactory working order, and results of subsequent runs were carefully checked by repeating the experiments.

In each series of impulse tests the steam pressure was increased by increments of 10 pounds up to 100 pounds gage pressure. The method of weighing the impulse proved to be very delicate, and the accuracy of the results is shown by the regularity with which they plot into smooth curves.

**Spacing of Buckets.**—The curve of bucket-spacing, Fig. 44, rises rapidly from zero, where the buckets are together and there is only lateral pressure, to 8.8 pounds for 100 pounds steam pressure and spacing from $\frac{5}{8}''$ to $\frac{1}{4}''$. The impulse then drops off gradually. The curve indicates that the spacing may vary from $\frac{1}{2}''$ to $\frac{7}{8}''$ without affecting the efficiency seriously; but apparently $\frac{5}{8}''$ to $\frac{3}{4}''$ pitch gives the greatest efficiency. This

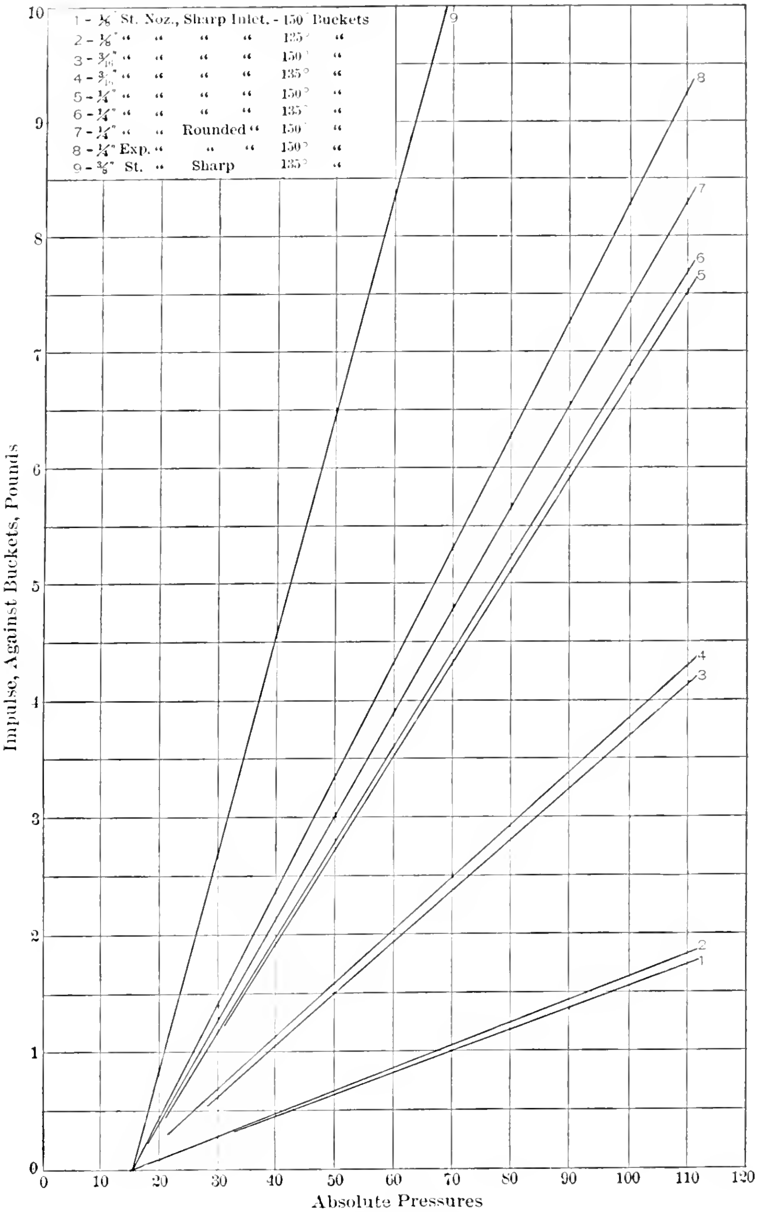



FIG. 42.—Curves showing impulse obtained with various steam pressures, using varying sizes of nozzle, and varying bucket angles. Upon the basis of these and the following curves, 135° buckets were decided upon for the experimental work.

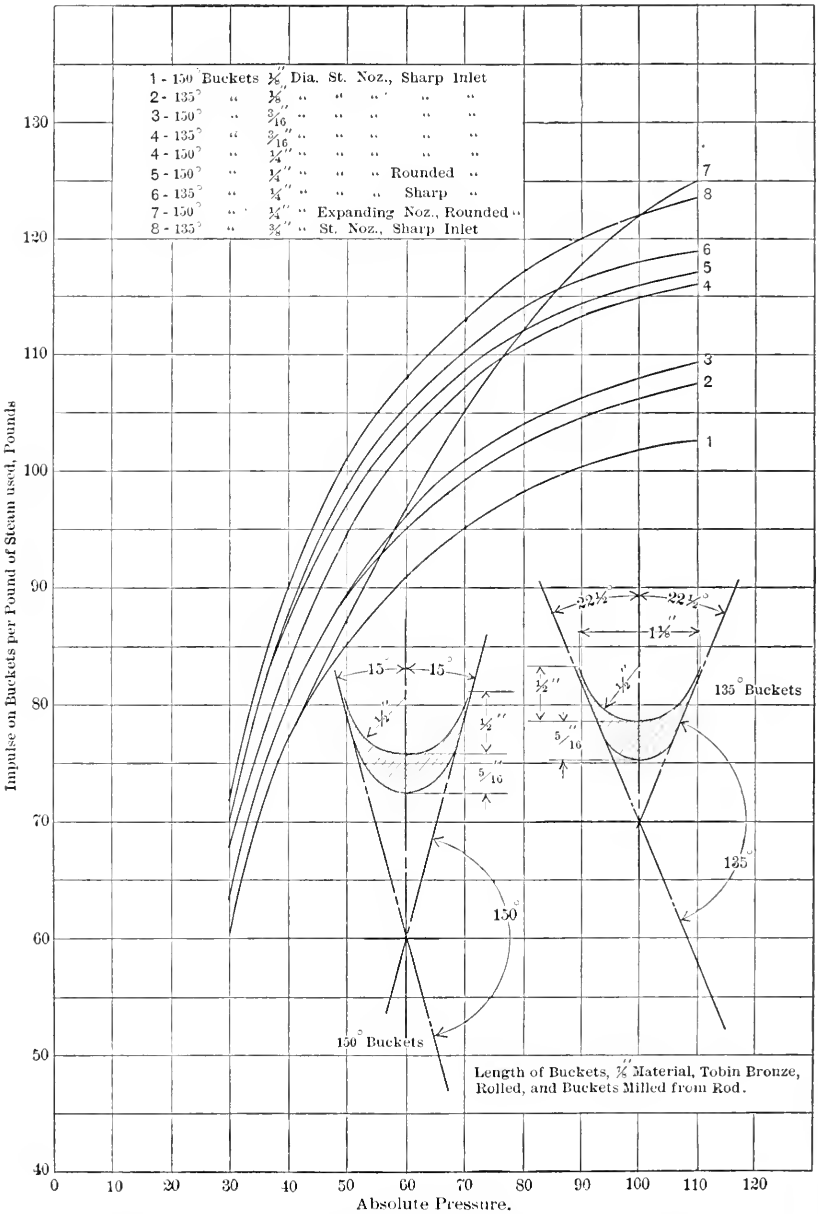
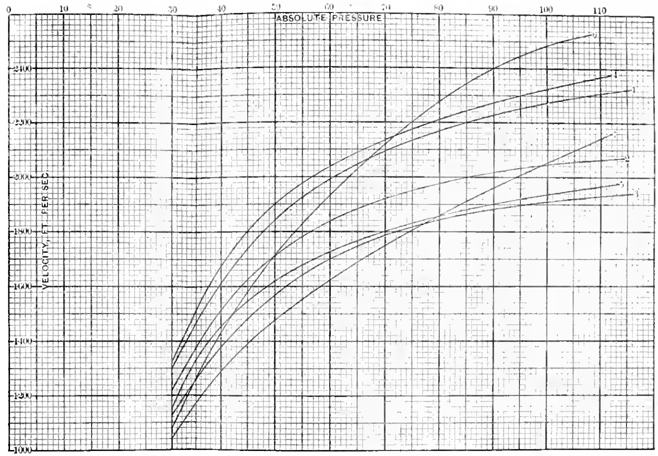



FIG. 43.—Impulse on buckets as produced by different nozzles. These curves express efficiency of buckets and nozzles together, in terms of impulse per pound of steam used.

## DATA FOR CURVES OF VELOCITY.

| Absolute Pressure. | Impulse, Pounds per Pound of Steam per Second. | Velocity Calculated from Impulse. | Velocity Calculated from Nozzle Reaction. |
|---|---|---|---|
| ¼″ Straight Nozzle, Sharp Entrance. 135° Buckets. | | | |
| 30 | 70.5 | 1228 | 1300 |
| 40 | 88 | 1532 | 1630 |
| 50 | 98.5 | 1715 | 1840 |
| 60 | 106 | 1845 | 2000 |
| 70 | 110.5 | 1920 | 2100 |
| 80 | 114 | 1982 | 2170 |
| 90 | 116.5 | 2025 | 2230 |
| 100 | 118 | 2055 | 2275 |
| 110 | 118.9 | 2065 | 2310 |
| ¼″ Straight Nozzle, Sharp Entrance. 150° Buckets. | | | |
| 30 | 68 | 1135 | 1300 |
| 40 | 83.5 | 1395 | 1630 |
| 50 | 94.5 | 1580 | 1840 |
| 60 | 102 | 1705 | 2000 |
| 70 | 107 | 1790 | 2100 |
| 80 | 111 | 1855 | 2170 |
| 90 | 113 | 1887 | 2230 |
| 100 | 115 | 1920 | 2275 |
| 110 | | | |
| ¼″ Straight Nozzle, Rounded Entrance. 150° Buckets. | | | |
| 30 | 70 | 1170 | 1330 |
| 40 | 87.5 | 1460 | 1700 |
| 50 | 97.2 | 1620 | 1900 |
| 60 | 103.7 | 1730 | 2040 |
| 70 | 108.5 | 1810 | 2150 |
| 80 | 112.2 | 1875 | 2230 |
| 90 | 114.2 | 1910 | 2280 |
| 100 | 116 | 1940 | 2320 |
| 110 | | | |
| ¼″ Expanding Nozzle, Rounded Entrance. 150° Buckets. | | | |
| 30 | 62 | 1035 | 1100 |
| 40 | 77 | 1285 | 1460 |
| 50 | 88 | 1470 | 1720 |
| 60 | 97 | 1620 | 1940 |
| 70 | 105 | 1750 | 2140 |
| 80 | 112 | 1870 | 2290 |
| 90 | 118 | 1970 | 2390 |
| 100 | 122 | 2040 | 2480 |
| 110 | | | |




¼″ Straight Nozzle, Sharp Entrance. . . . . { Curve 1, Velocity from Reaction.
" 2, " " Impulse, on 135° Buckets.
" 3, " " " " 150° "

¼″ Straight Nozzle, Rounded Entrance . . . { " 4, " " Reaction.
" 5, " " Impulse, 150° Buckets.

¼″ Expanding Nozzle, Rounded Entrance { " 6, " " Reaction.
" 7, " " Impulse, 150° Buckets.

*Note.*—These curves show comparative values of the velocity of steam-jets as calculated from the measured reaction against the nozzle and attachments, and from the impulse exerted by the jets upon the buckets shown in Fig. 43. The losses due to friction, eddies, etc., in the buckets, cause the velocity as indicated by the impulse to be less than that indicated by the reaction. Curves 6 and 7, from the expanding nozzle, show the same characteristics as noted at bottom of page 125, and as shown also by Curve 7, Fig. 43.

[*To face page* 128.

being a convenient spacing from constructive considerations, it was adopted for the subsequent experiments.

**Effect of Clearance between the Nozzle and the Buckets.**—By means of shims between the nozzle support and the clamp

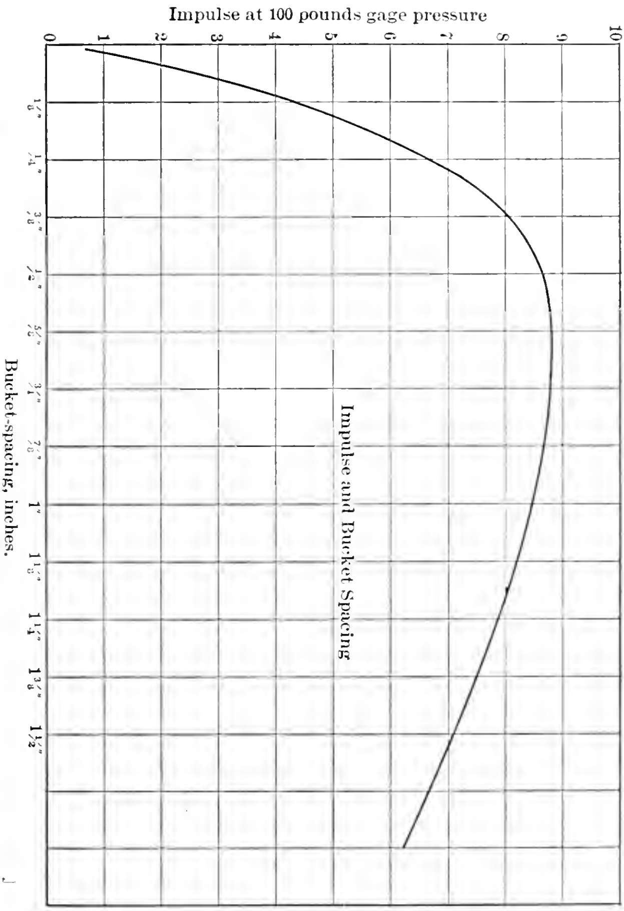



Fig. 44.—These results led to the adoption of $\frac{3}{4}''$ spacing in the experiments.

carrying the buckets, the effect of placing the nozzle end at varying distances from the buckets was tested, the distances varying from $\frac{15}{32}''$ to $\frac{23}{32}''$. Very little difference in impulse could be detected, and only a few points were found, as shown

in Fig. 45. Apparently within the limits used, the distance of the nozzle from the buckets is not of great importance.

**Effect of Additional Sets of Buckets,** through which the steam passes on its way to the movable buckets.

With one set of stationary nozzles clamped in front of the movable buckets (these being reversed in position and the scales counter-weighted so as to measure the impulse), at 100 pounds per square inch gage pressure, the impulse on the movable buckets was 6 pounds. With two stationary sets clamped together without clearance between them, and placed before the movable nozzles as before, the impulse on the movable buckets was 4.8 lbs. With three sets of stationary buckets the impulse was 3.6 pounds. When no extra sets of buckets were used, the impulse on the movable buckets due to the direct jet from the nozzle was 8.8 pounds for an initial pressure of 100 pounds gage.

If with two extra sets the first set of extra buckets (stationary) should receive 8.8 pounds, the second set 6, and the movable 4.8 pounds, the total impulse would be the sum of 8.8, 6.0, and 4.8, or 19.6 pounds. The upper curve (Fig. 46) was plotted upon this assumption, adding to the impulse of the first set that of all the following. It has been the experience of builders of the many-stage impulse-turbine that the pressure beyond a row of buckets is often higher than that before it, and it is probable that in the arrangement under discussion the steam-flow would be checked by the accumulation of pressure in the later buckets, thus preventing the full impulse from being realized.

The middle curve shows the obtainable impulse for the ordinary arrangement of impulse-turbine, in which only the alternate rows of buckets rotate, the others being the stationary guides. The total impulse given by this arrangement is much greater than that given by the single row of buckets, but not as great as though all the rows rotated.

While these curves indicate relative values of the losses occurring in the guide-blades, the results are probably quite different, numerically, when the movable buckets are travelling

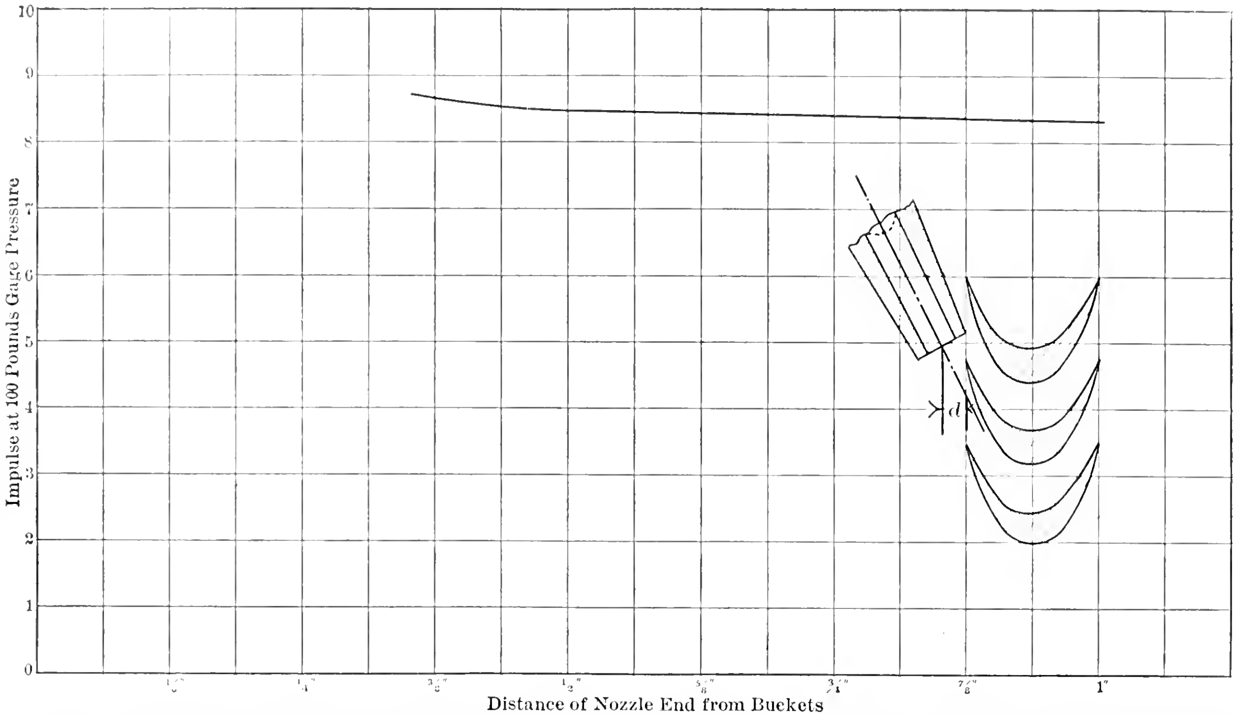



FIG. 45.—The distances $d$ refer to the distance of the center of the exit end of the nozzle from the inlet edge of the buckets. The plane of the nozzle end was square with the nozzle axis, and not beveled off as is ordinarily done in impulse-turbine practice.

rapidly in front of the guide-buckets and disturbing the steady flow of steam.

**Effect of Clearance between Sets of Buckets.**—In turbine construction it is necessary to provide clearance between the

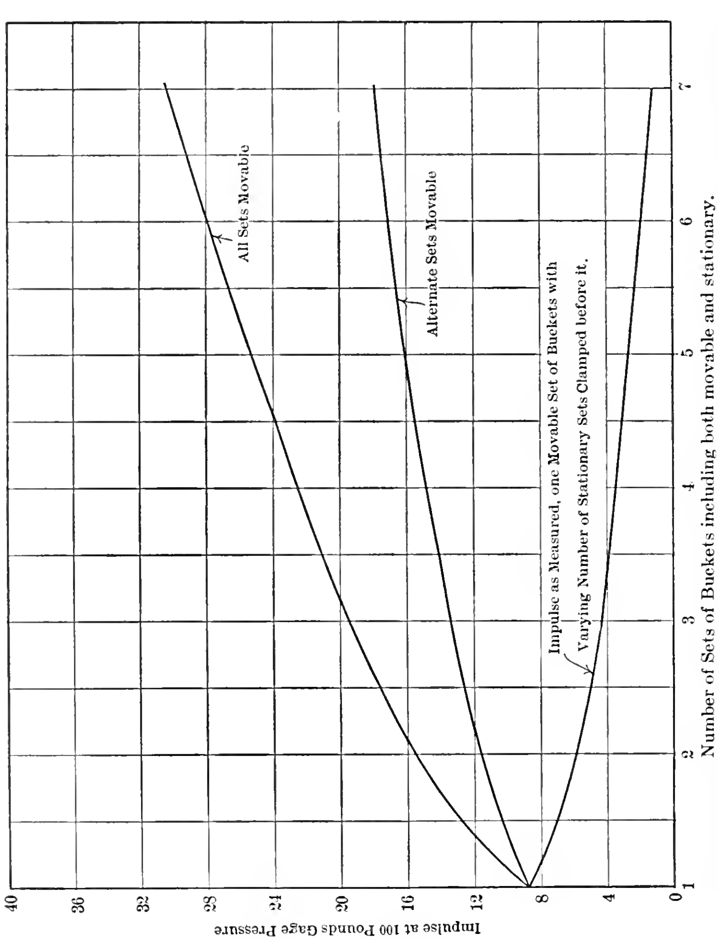



FIG. 46.—Effect of additional sets of buckets. Lower curve shows the loss of impulse due to the steam having to pass through a varying number of rows of guide-buckets before reaching the moving buckets.

moving and stationary rows of blades or buckets, and this was not allowed in the previously described experiment for finding the effect upon the impulse of increasing the number of rows of stationary buckets.

To determine the effect of clearance two sets of stationary buckets were placed before the movable set, and the clearance was obtained by interposing strips of sheet metal between the stationary sets. Runs were made with clearances of $\frac{1}{32}''$, $\frac{1}{16}''$, and $\frac{1}{8}''$.

The curve at the top of Fig. 47 shows the impulse at 80 pounds initial pressure with varying amounts of clearance. The points determined all fall on a smooth curve, and show that clearance up to $\frac{1}{32}''$ has apparently very little effect in diminishing impulse. From $\frac{1}{32}''$ to $\frac{1}{16}''$ the loss is noticeable, and after $\frac{1}{16}''$ it is great, increasing rapidly with the clearance. On the lower part of the page are shown curves of impulse with different clearances. Calling the impulse obtained with no clearance at all 100 per cent, the losses due to increased clearance are as follows at 100 pds. initial pressure by gage.

| | | | | |
|---|---|---|---|---|
| Buckets clamped close together, no clearance ........................ | impulse | 4.8 | pds. | = 100% |
| $\frac{1}{32}$-inch clearance.................. | " | 4.8 | " | = 100% |
| $\frac{1}{16}$- " " .................. | " | 4.5 | " | = 94% |
| - " " .................. | " | 3.6 | " | = 75% |

These figures and the curves indicate that the clearance between rows has an important bearing upon turbine economy. A certain amount of clearance is necessary for mechanical reasons, especially since the parts of the machine are exposed to high temperatures. Especial attention to this point is required in machines that are to use superheated steam.

**Use of Air instead of Steam.**—The nozzle directing the jet upon the buckets was attached to a source of compressed-air supply, the remainder of the apparatus being the same as that used in the steam experiments excepting that the canvas shield used with steam was no longer necessary.

As is shown by the curves (Figs. 48 and 50), the impulse with air was in each case about 12 per cent higher than with steam of corresponding initial pressure. The effects produced

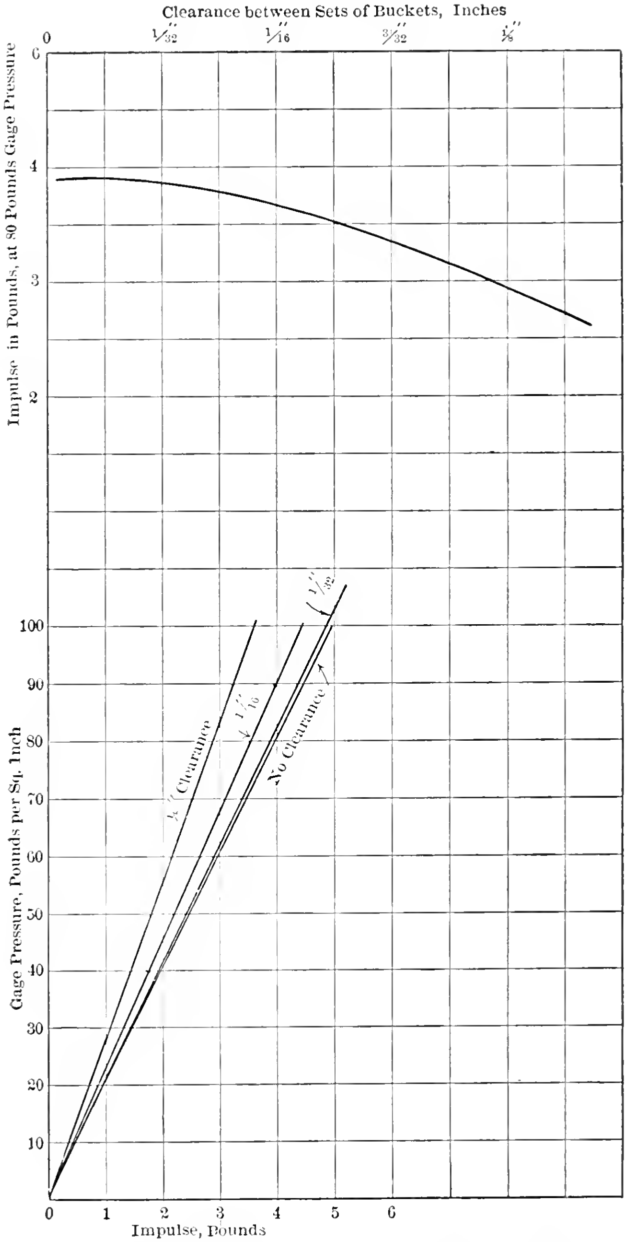



FIG 47.—Relation between impulse and clearance between buckets, showing decrease of the impulse due to increase of clearance.

were the same in character as those produced by steam, and as air was more agreeable to operate, the remaining experiments were made with it instead of steam.

**Effect of "Cutting Over" the Edges of the Buckets.**—The nozzle angle was shifted from its former position so that instead of directing the jet tangentially upon the bucket surfaces at entrance, it caused the stream to be divided or split by the

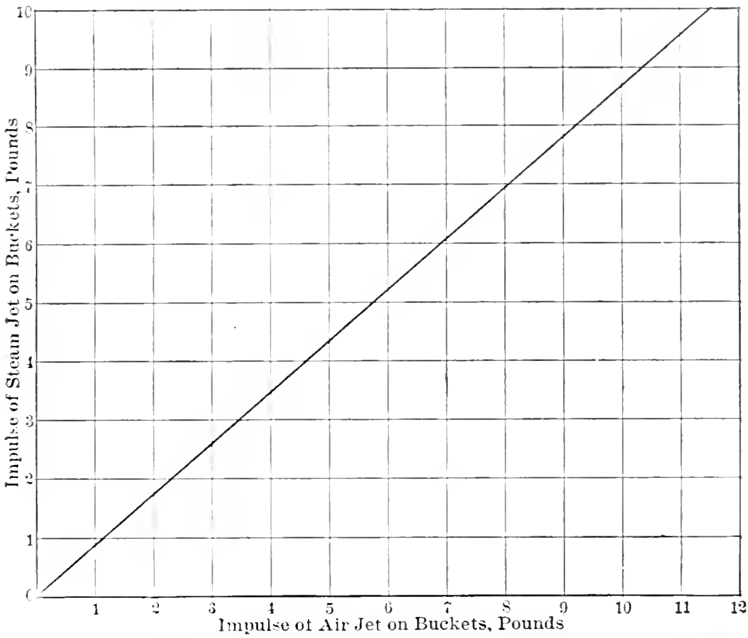



FIG. 48.—Relation between impulse produced by steam and by air.

edges. The results are shown in Fig. 49 for 100 pounds initial pressure. The most efficient angle was found to be that given tangency of the stream to the buckets, or $22\frac{1}{2}$ degrees with the vertical. Larger angles cause an action against the backs of the buckets, while with smaller angles the stream is spread by the edges of a number of buckets and does not strike any as efficiently as when directed tangentially to the bucket surfaces at entrance.

**Efficiency of Rough Surface Buckets as Compared with those having Smooth Surfaces.**—The buckets as used in the previous experiments had been finished to very smooth surfaces and it was desired to find out to what extent this contributed towards high efficiency. The buckets were therefore taken from the clamps, covered with shellac and sprinkled with

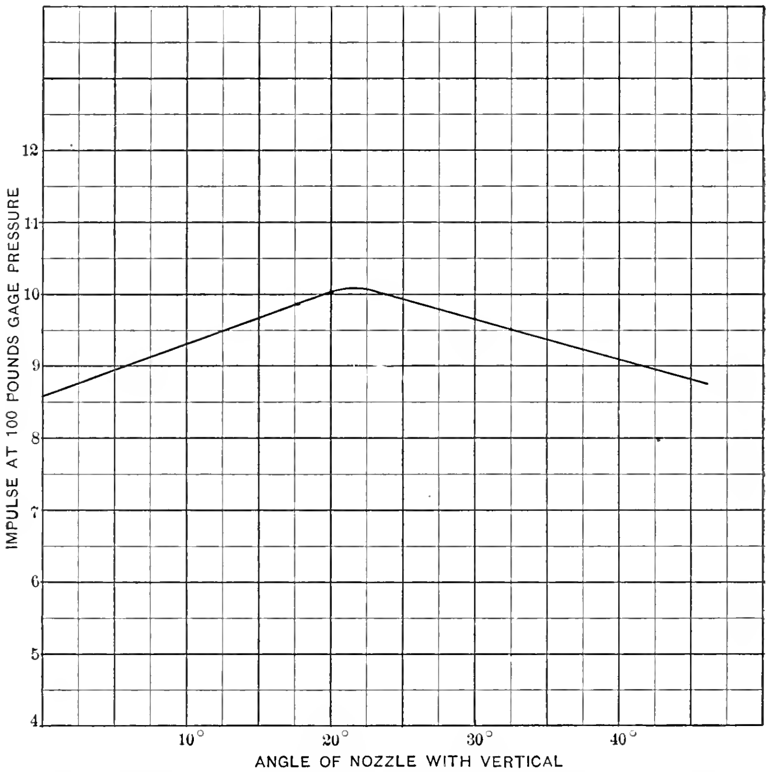



FIG. 49.—Effect of cutting over the edges of the buckets. For these experiments $22\frac{1}{2}°$ was found to be the angle of nozzles giving highest efficiency.

brass filings. These were allowed to stick and they effectually roughened the surfaces. The buckets were then reset in the clamps and runs were made, using air as the working fluid, with one set of movable buckets and also with two stationary sets placed before the movable set.

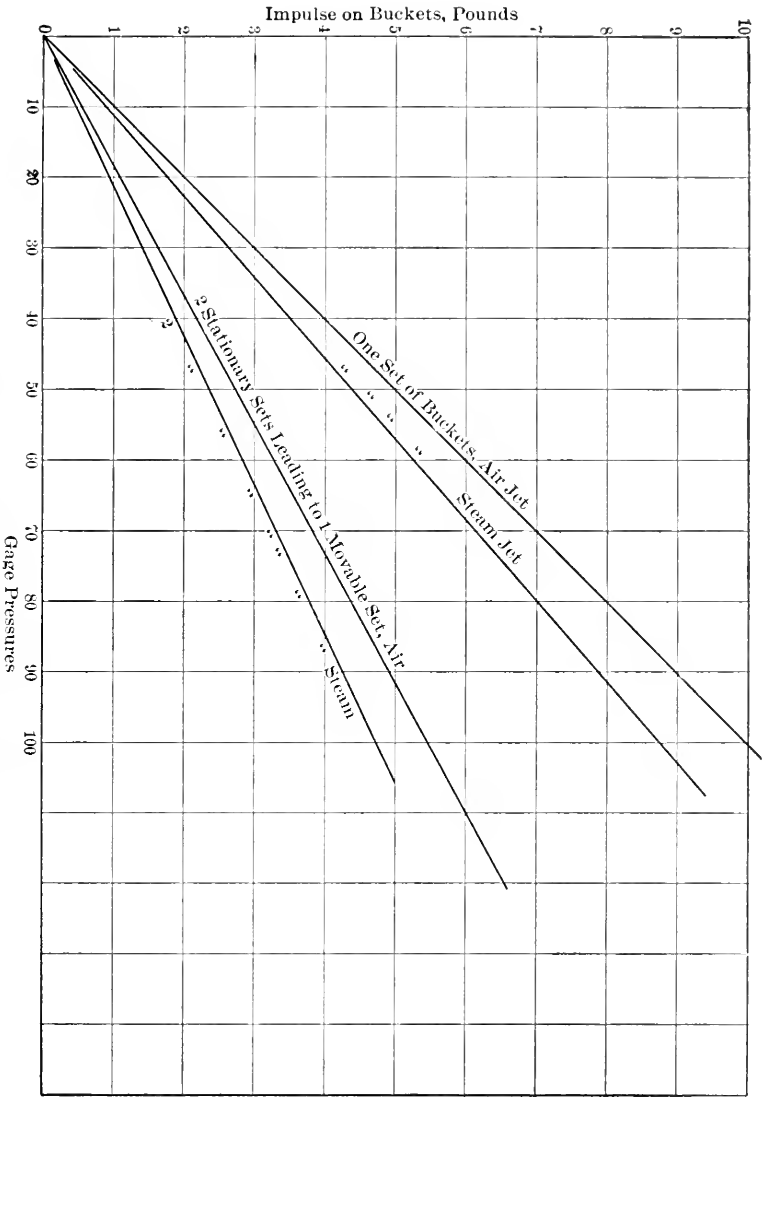



Fig. 50.—Relative impulse, steam and air jets, of smooth buckets.

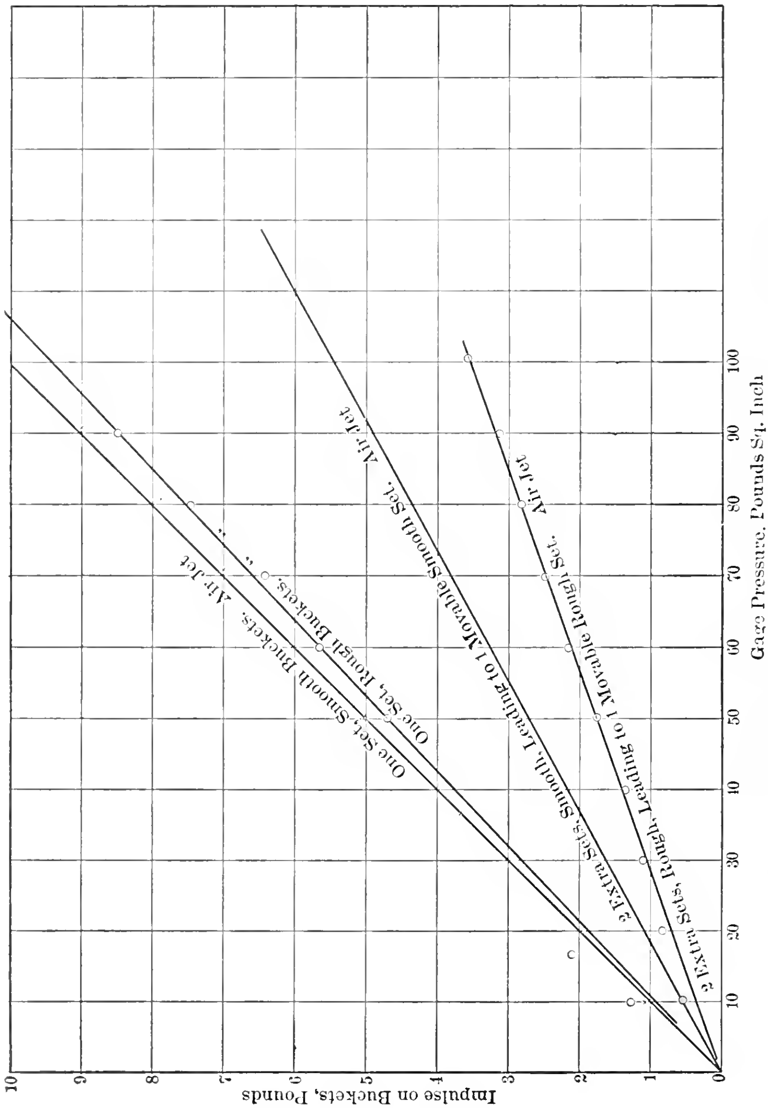



FIG. 51.—Relative effective impulse with rough and smooth buckets.

The losses resulting from increased skin friction were very considerable. With one set of movable buckets only, the loss amounted to 6 per cent—that is, the impulse at 100 pounds initial gage pressure was only 94 per cent of the impulse for smooth buckets at the same pressure. The curves, Fig. 51, show the relation between the impulse as received upon smooth and upon rough buckets respectively. The runs made with two extra sets of rough buckets placed before the set of movable buckets show very much increased losses and indicate that the loss is directly proportional to the number of sets added. The investigators plotted a curve (not reproduced here) based on this indication, and concluded that, calling the smooth buckets 100 per cent efficient, the following would result from the addition of successive sets of rough buckets of the kind employed in the experiments.

| | Efficiency. |
|---|---|
| One set smooth buckets. . . . . . . . . . . . . . | 100 per cent. |
| " " rough " . . . . . . . . . . . . . . . | 94 " " |
| Two sets rough " . . . . . . . . . . . . . . . | 82 " " |
| Three " " " . . . . . . . . . . . . . . . | 64 " " |
| Four " " " . . . . . . . . . . . . . . . | 42 " " |

This means that if the working fluid were caused to pass through four sets of such rough buckets as used, before striking the single movable row of rough buckets, the impulse upon the latter would be less than half of what would be obtained with one set of smooth buckets acted upon directly by the jet from the nozzle.

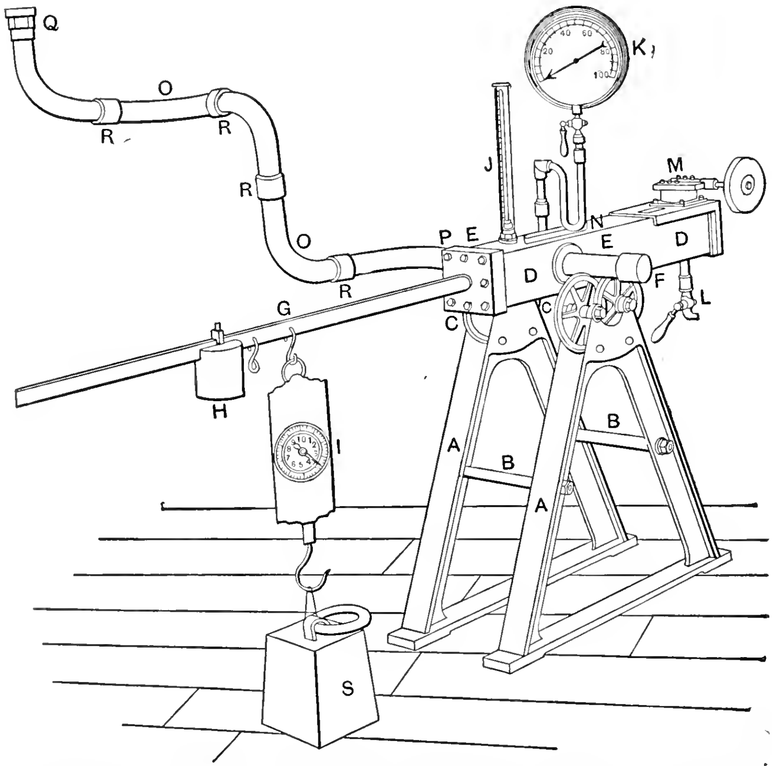
**The following inferences are drawn from the experimental work discussed in the preceding pages:**

1. Rate of flow, by weight, is greater through an orifice with rounded entrance than if the entrance is sharp-cornered or only slightly rounded.

2. Rate of flow, by weight, is decreased by the addition of a nozzle, either diverging or straight, to the discharge side of the orifice.

3. Rate of flow, by weight, reaches a maximum when the final pressure is from about 0.85 to 0.50 times the absolute initial pressure.

4. The maximum rate of flow from the sharp-cornered orifice occurs after a somewhat greater reduction of back pres-



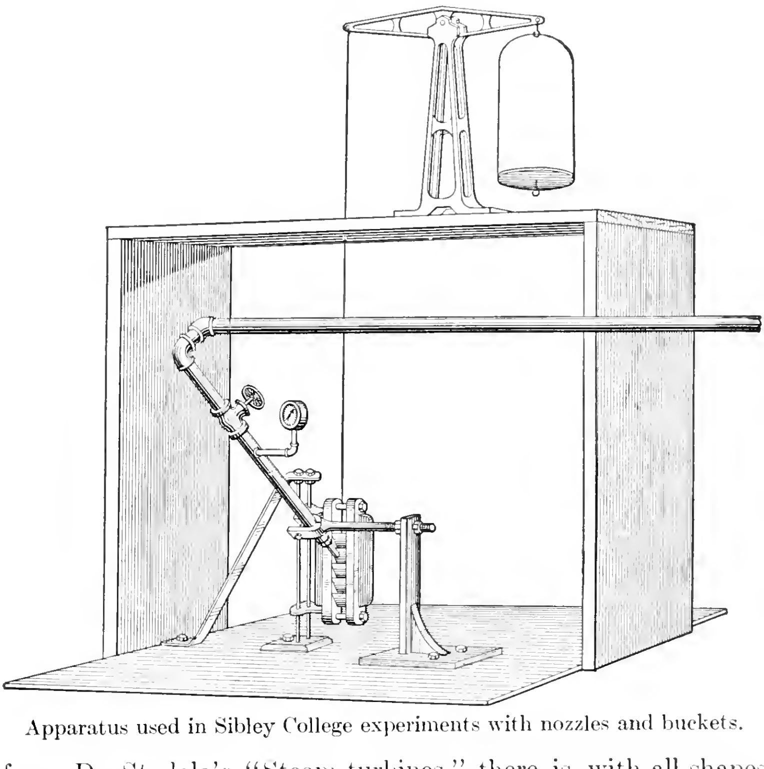

Apparatus used by Mr. George Wilson, for determining reaction due to steam flow from orifice at *M*. (Reproduced from London "Engineering," 1872.)

sure than is required with the rounded orifice to bring about the maximum rate of flow.

5. The addition of a divergent nozzle to the orifice seems to cause the maximum rate of flow to occur earlier—that is, after less reduction of back pressure—than is the case with the simple orifice.

6. The velocity attained depends to some extent upon the rounding of the orifice or entrance to the nozzle, and may be greater with the square or slightly rounded entrance than when the rounding is of greater radius.

7. As shown in Figs. 53 and 54, from the experiments of Messrs. Weber and Law in Sibley College, and Fig. 55,



Apparatus used in Sibley College experiments with nozzles and buckets.

from Dr. Stodola's "Steam-turbines," there is, with all shapes of orifice there represented, a sudden drop of pressure immediately in the narrowest section of the orifice, to below the back pressure, then a rise of pressure as the steam leaves the orifice, accompanied by variations above and below the back pressure, till the pressure in the jet gradually steadies down to that of the medium into which it is flowing. The Sibley College experiments were made with the searching-tube

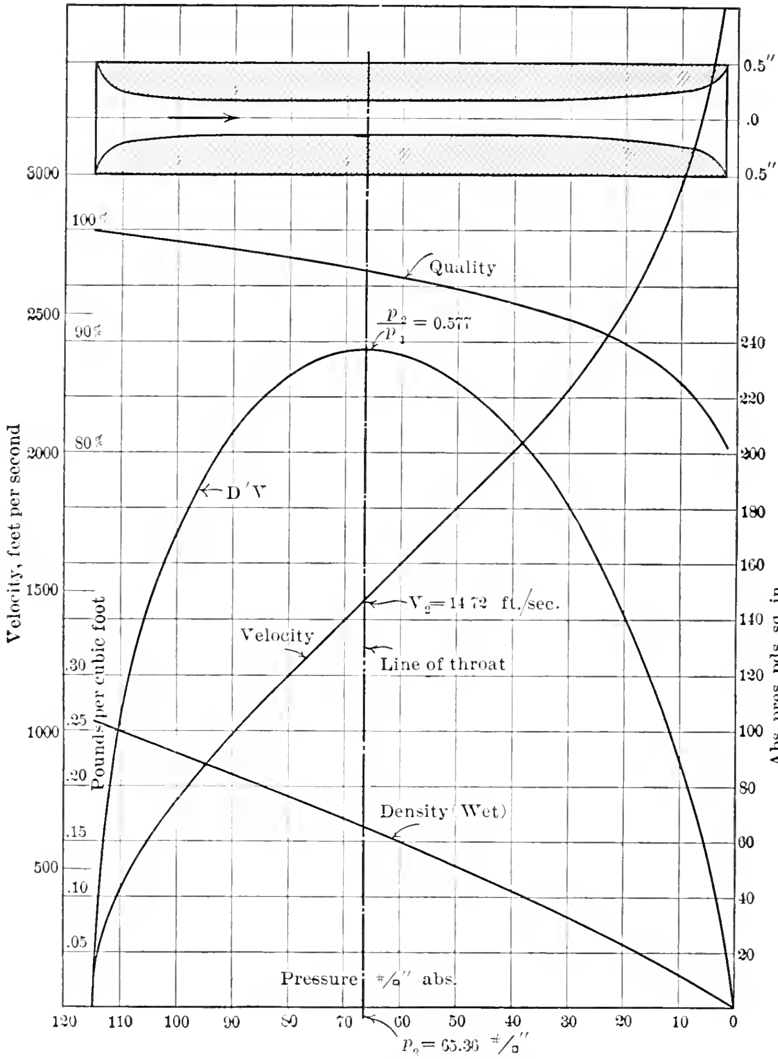



FIG. 52.—Curves representing ideal conditions of flow, with adiabatic expansion, and nozzle cross-sections made so as to carry out such expansion

communicating with the piston of an ordinary steam-engine indicator, and the rapid vibrations were not indicated to the same extent as in the experiments described by Dr. Stodola.

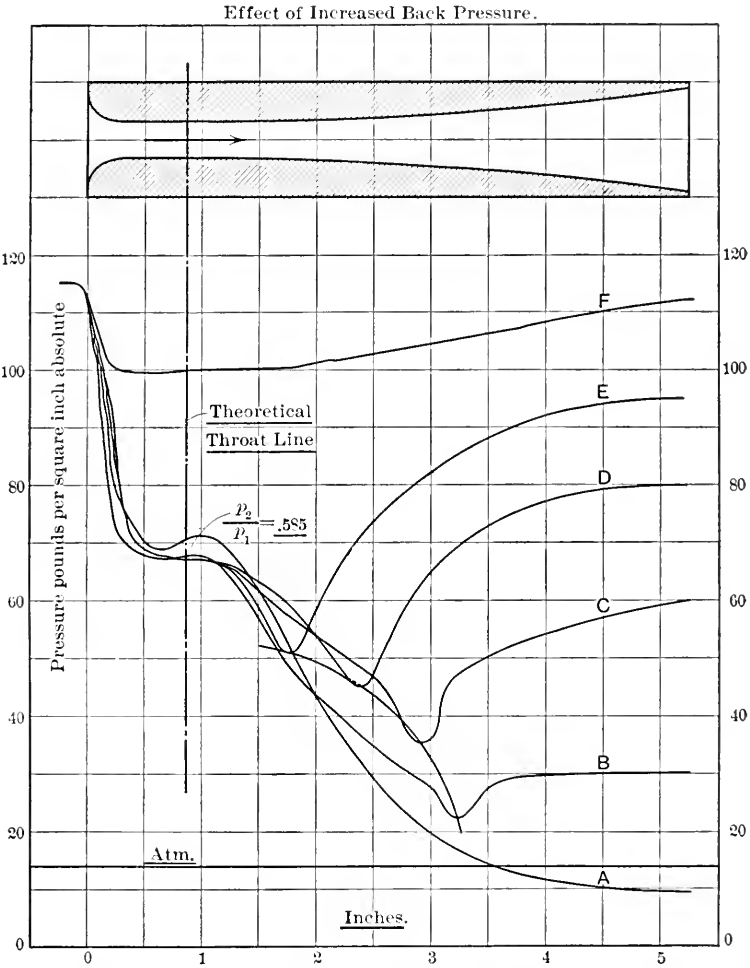



FIG. 53.—Nozzle used in experiments of Messrs. Weber and Law, and curves obtained with varying back pressures.

8. According to the experimental work discussed, a simple orifice is more efficient than an expanding nozzle for initial

pressure up to about 70 pounds absolute; for higher initial pressures an expanding nozzle, with entrance only slightly rounded, is to be used, and its efficiency increases as the initial

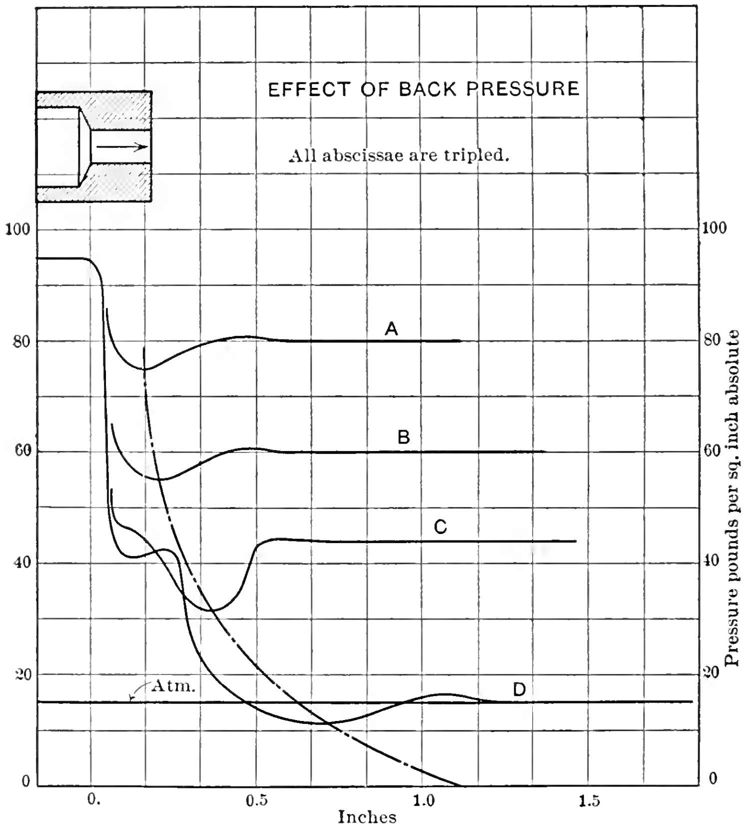



FIG. 54.—Orifice used by Messrs. Weber and Law, and curves obtained with varying back pressures.

pressure increases. Plate III shows a value of $y=0.06$ for pressures about 200 pounds by gage. Such high efficiency cannot be obtained with an incorrectly designed nozzle.

9. It appears that steam flowing through a simple orifice does not attain a greater velocity, *while in the orifice itself*, than from 1400 to 1500 feet per second, no matter how much the

pressure is reduced in the receiving space. However, as shown on page 117, the velocity of the jet issuing from a simple orifice into the atmosphere, as indicated by the reaction against the discharging vessel, may be as high as from 2600 to 2700 feet per second. The fact that the weight of flow

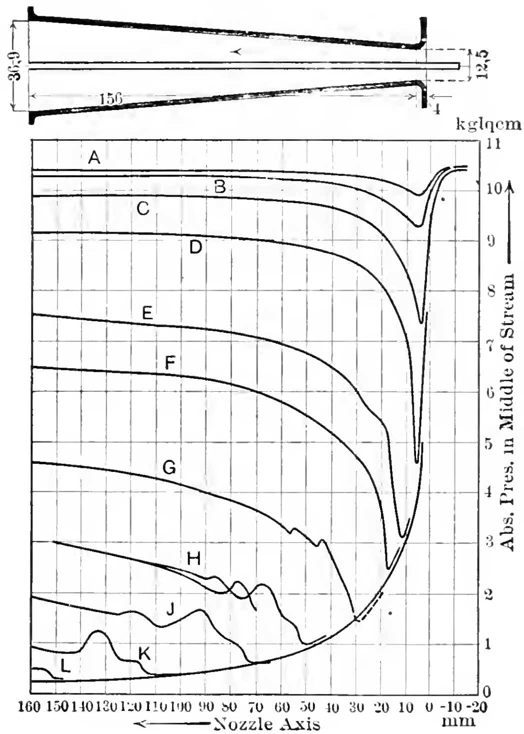



FIG. 55.*

can be so closely calculated upon the basis of a heat drop corresponding to about 1500 feet per second velocity in the orifice is significant of the truth of the first statement. The additional circumstances that the reaction indicates a much greater final velocity of efflux, and that the simple orifice has been found in practice to be superior in efficiency to the expanding nozzle

* Figs. 55, 55*a*, and 56 are from Dr. Stodola's book on Steam Turbines.

for low initial pressures, lead to the conclusion that a considerable portion of the energy in the steam after it leaves the throat is effective in further accelerating the jet in its initial direction. The remainder of the energy given up is spent in producing the vibrations already described, and in causing a general displacement of the atmosphere into which the jet flows. It is the province of the expanding nozzle attached to the simple orifice to contain the steam during its total expansion from initial to lowest possible back pressure, and to thus cause the velocity of the jet to attain the maximum value corresponding to the total change from energy in the form of heat to kinetic energy of the jet, and to direct the flow into a given line of action, so that the jet may be usefully employed.

10. In the divergent or expanding nozzle the interchange of heat energy between the steam and the walls of the nozzle causes more heat to be rejected in the exhaust than would be rejected if the flow were frictionless. This is one cause of loss of energy and therefore of diminished efficiency.

11. Another loss of energy may occur, due to incorrect proportions of the nozzle; that is, while having correct cross-sectional areas for the desired flow of steam, the nozzle may be too long or too short, and thus the angle of divergence may be such that the jet will leave the nozzle walls and so not fill out the cross-sections. This leads to vibrations of the stream and consequent loss of energy. The nozzle should be so arranged that the steam will expand while in the nozzle to just the pressure of the medium into which it is to flow. The curves *A*, *C*, and *D*, in Fig. 56, show the vibrations occurring when the back pressure is either less or greater than that at the end of expansion in the nozzle. Curve *B* shows the correct condition, the back pressure being just that at the large end of the nozzle. In Figs. 53 and 54 are shown curves obtained by Messrs. Weber and Law by the use of a searching-tube and indicator as before described.

These curves show, for varying back pressures but con-

stant initial pressure, the drop occurring at once upon arrival of the steam in the throat of the nozzle, and the rise following the initial drop of pressure. The smooth curve bounding

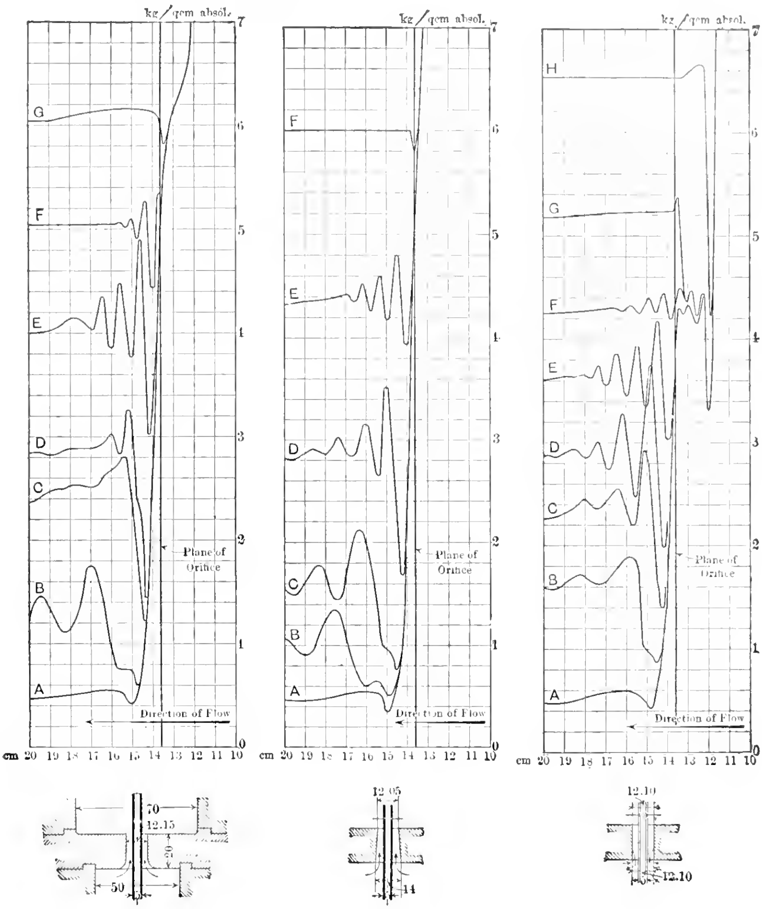



FIG. 55*a*.

the ends of the pressure curves on Fig. 55 is the curve of adiabatic expansion.

The fact seems to be that a great increase in velocity occurs at entrance to the nozzle, after which the velocity is checked and the pressure rises. Dr. Stodola explains this as ". . . be-

cause steam particles possessed of great velocity strike against a slower-moving steam mass, and are therefore compressed to a higher degree . . . according to the theory of 'compression shock' of Von Riemann."

Curve *N*, Plate IX, was plotted from a tabulated series of results of experiments published by Dr. Stodola in his work, "The Steam-turbine." The curve represents the fall in pressure as the steam advanced along the nozzle shown above the curves; curve *A* has been calculated with the value $y=0.20$.

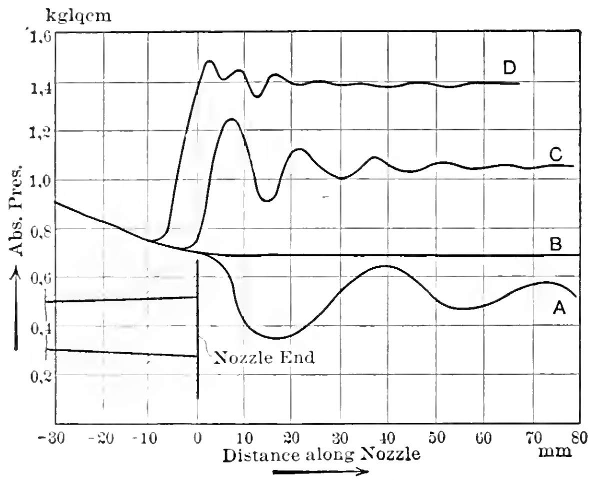


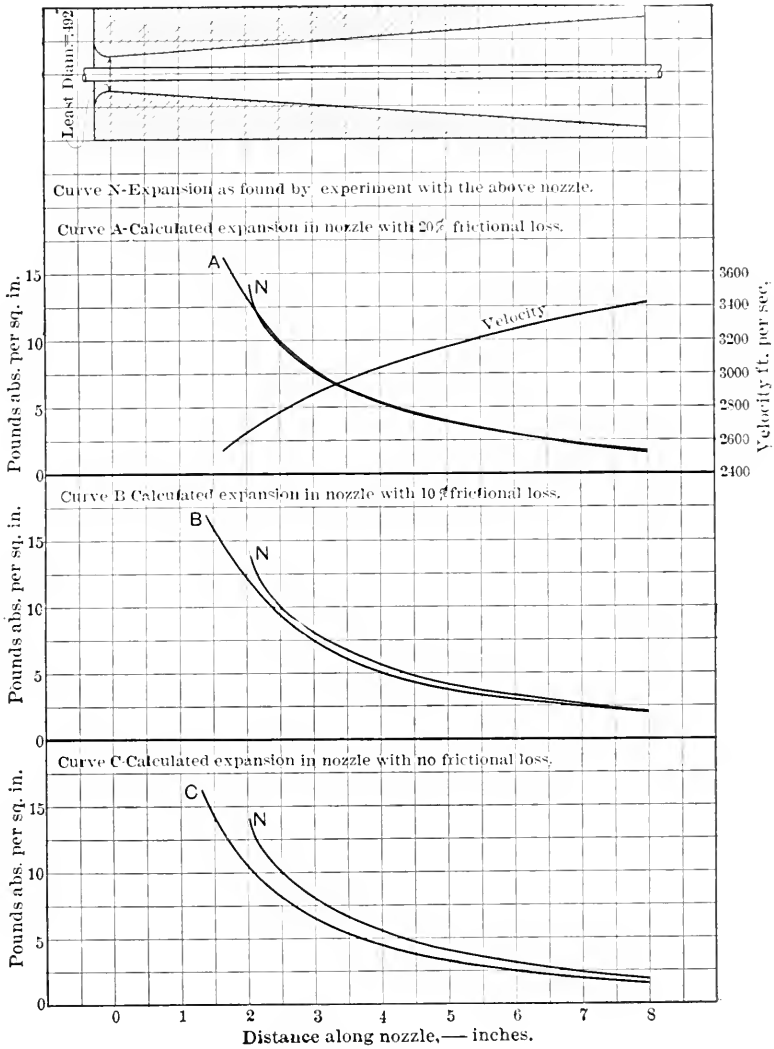

Fig. 56.

This is seen to coincide very closely with the experimentally determined curve.

Curves *B* and *C* represent calculated pressures along the nozzle with allowances of 10% and 0 energy loss respectively. Assuming that the nozzle was so designed that the steam filled out the cross-sectional areas, the velocities along the nozzle were as given by the velocity curve, reaching about 3400 feet per second.

The experimentally determined pressures used in plotting curve *N* were those obtained with the hole in the side of the searching-tube sloping against the stream, and were higher

PLATE IX.


Least Diam.=.492
Curve N-Expansion as found by experiment with the above nozzle.
Curve A-Calculated expansion in nozzle with 20% frictional loss.
A
N
Velocity
Pounds abs. per sq. in.
0
5
10
15
Velocity ft. per sec.
2400
2600
2800
3000
3200
3400
3600
Curve B Calculated expansion in nozzle with 10% frictional loss.
B
N
Pounds abs. per sq. in.
0
5
10
15
Curve C-Calculated expansion in nozzle with no frictional loss.
C
N
Pounds abs. per sq. in.
0
5
10
15
0
1
2
3
4
5
6
7
8
Distance along nozzle,— inches.

than when the hole was normal, or when it sloped away from the stream. If the lower values are more nearly correct, then the energy loss was less than 20%.

The initial pressure used in the experiments is given as 149 pounds absolute. By comparing the friction loss of 20% with that indicated on Plate III, the latter is, for the same initial pressure, only 12%, and this tends to confirm the inference pointed out by Dr. Stodola, that the friction loss is lower than 20%. The values of $y$ calculated in the table at the end of Chapter V, from the Sibley College experiments, show, for 110 pounds absolute pressure, a frictional loss of 12.5% in the expanding nozzle used.

# CHAPTER VII.

## THE IMPULSE-TURBINE.

THE impulse-turbine may be designed in any one of the following ways:

(*a*) Single stage, consisting of a set of nozzles and a single wheel carrying one row of blades. The pressure is the same on the two sides of the wheel, or disc, the whole pressure drop occurring in the nozzles. This gives very high peripheral velocity, and since the diameter must be kept small enough to keep frictional resistances within limits, the number of revolutions is very great. The de Laval turbines run at speeds of from 10,000 to 30,000 revolutions per minute, giving a peripheral velocity of 1200 to 1400 ft. per second. The excessive angular velocity of the rotating part necessitates the use of gearing in applying the power to machines.

(*b*) Other rows of blades may be added, either upon the single wheel or upon separate wheels, in order more completely to absorb the energy of the steam leaving the nozzles. There is no further pressure drop, however, after leaving the nozzles, and only one set of the latter is supplied. This type has therefore a single pressure stage and several velocity stages.

(*c*) The first nozzles may be so arranged as to expand the steam through only a portion of the pressure and temperature range available, thus causing the steam to leave the first set of nozzles at a much lower velocity than results from the single-

pressure-stage turbine. Since good efficiency demands that the peripheral velocity of the blades be proportional to the entering steam velocity, the peripheral velocity may be decreased with the decrease of steam velocity. The steam is reduced in the first nozzles to a pressure considerably higher than the condenser pressure, and hence may be expanded through another set of nozzles arranged to discharge upon another set of blades, on a separate wheel, in a separate compartment or division of the turbine-casing from that containing the first wheel. The second set of nozzles and blades constitutes the second stage of the turbine. By sufficiently limiting the pressure drop that can occur in a single set of nozzles, the velocity of exit of the steam, and consequently the necessary peripheral velocity of the blades, may be greatly reduced. The many-stage impulse-turbine thus consists of several single-stage turbines, placed in series with one another. The steam leaves each set of blades with considerable velocity, but since the next wheel is in a separate chamber, and the steam has to pass through a set of orifices or nozzles to reach it, the exit velocity cannot be used as velocity. The steam comes partially to rest before going through the next nozzles, and the energy in the exhaust from the preceding blades is expended in producing impact, and consequently in raising the temperature and pressure of the steam before it enters the succeeding nozzles. Thus the exit velocity from all but the wheel next to the condenser is effective in doing work in the turbine. In passing through the chambers and passages there is loss due to leakage through the clearance spaces, and this causes loss of the heat in a certain amount of steam which gets through without doing work on the turbine-buckets.

**The Single-stage Impulse-turbine.**—The velocity of steam at exit from a nozzle may be determined as previously indicated, and gives the value shown by $V$ in Fig. 57, being the absolute velocity of the steam as it enters the turbine.

Considering first a simple impulse-wheel, rotating with a peripheral velocity of $u$ feet per second, the velocity of the

entering steam, relatively to the velocity of the rotating blades on the wheel, will be represented by $v$ ($=AC$) in magnitude and direction. In order that the steam may enter the blades without shock, the angle of the entering edge of the blades with the direction of motion, $u$, must be $\Delta$, the same as the direction of relative velocity of the entering steam. Assuming that no frictional losses occur in the blade-channels, the relative exit velocity will be $v_1 = v$. The angle of exit may be made according to the judgment of the designer, and, as has been

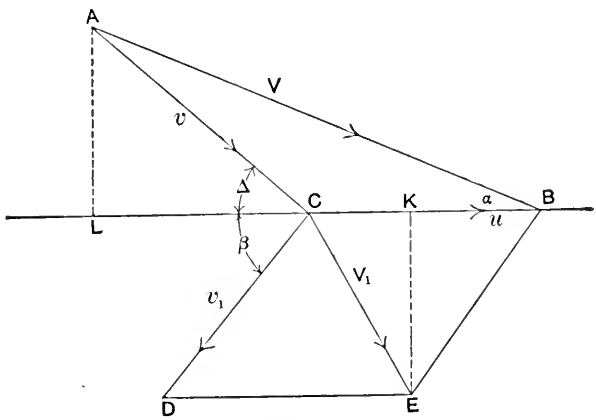



FIG. 57.

seen (see page 20), this angle determines to a great extent the efficiency of the wheel. Mechanical considerations prevent the obtaining of complete reversal of the jet in this type of turbine. Usually the angle $\beta$ is made equal to the angle $\Delta$, and the cross-sectional area at exit from the blades equals that at entrance to them.

It is shown by the examples on page 23 that if $V$ and $V_1$ are, respectively, the absolute velocities of the entering and departing steam, the work done upon the blades by $W$ pounds of steam passing them per second is

$$K = W(V^2 - V_1^2) \div 2g, \text{ foot-pounds.}$$

Since the kinetic energy at velocity $V=\frac{WV^2}{2g}$, the efficiency is $\frac{V^2-V_1^2}{V^2}$.

The velocities may be represented as shown in Fig. 58, $V$ and $V_1$ being the initial and final absolute velocities respectively.

Let the initial velocity be 3500 feet per second, $=V$.

" $\alpha=30°$.

" peripheral velocity $=1200$ feet per second, $=u$. For the ideal case shown at the left on Plate X the relative entrance and exit velocity is $v=2540$ ft. per sec. This gives $V_1$, the absolute exit velocity, as 1870 ft. per sec. The energy given up to the buckets, per pound of steam, is

$$\frac{(3500)^2-(1870)^2}{64.4}=136,000 \text{ foot-pounds.}$$

This may also be computed by resolving the absolute velocities $V$ and $V_1$ along the direction of motion of the buckets, and adding the components, multiplying by the peripheral velocity, $u$, and dividing by $g$. The horizontal components may be taken from the diagram by measurement.

Thus the energy given up is

$$\frac{(C+C_1)u}{g}=\frac{(3030+640)\times 1200}{32.2}=136,000+.$$

**Losses in Nozzles and Buckets.**—As the steam expands in the nozzle it experiences frictional resistances which cause it to give up less energy than it would under ideal conditions of flow, and the loss therefrom diminishes the nozzle exit velocity, $V$, to some value $fV$ $(=V')$, where $f$ is equal to the square root of the quantity $1-y$ in the example on page 83. Thus, for $y=0.15$, $f=\sqrt{0.85}=0.92$.

The coefficient $f$ varies according to the length and other

PLATE X.

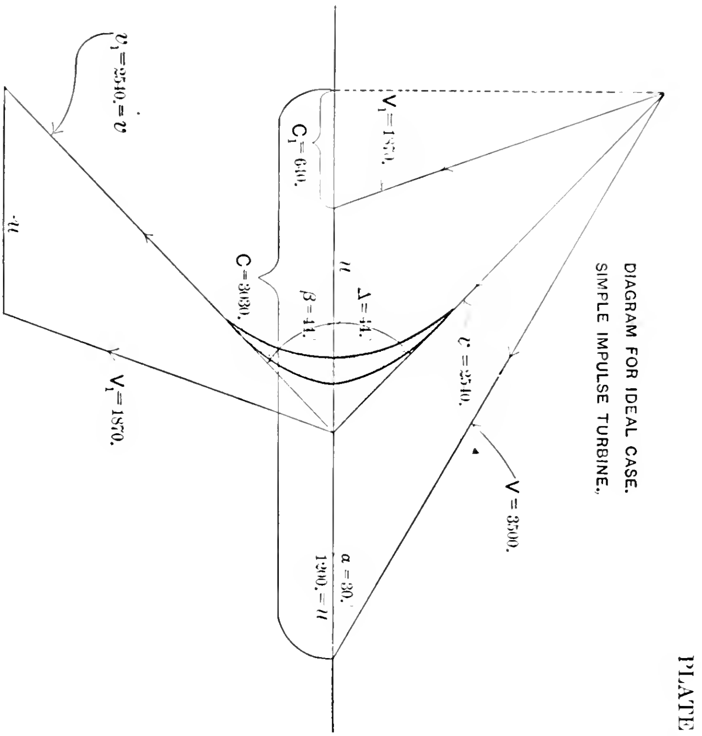



FIG. 58.

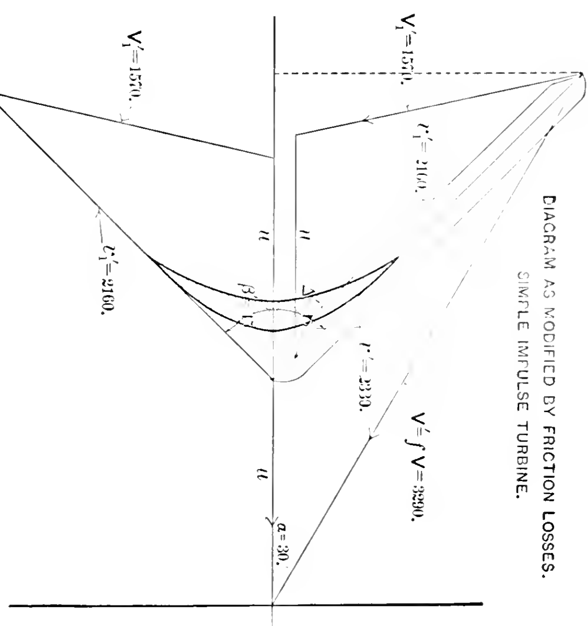



FIG. 59

proportions of the nozzle. The initial velocity being $V'$ $(=fV)$ gives $v'$ as the real relative velocity of the steam at entrance to the first moving blades of a stage. This is further decreased, by resistances in the blades, to the value $v_1'=kv'$. The loss of energy, per pound of steam, will be, in the nozzle,

$$L_N=\frac{V^2-(fV)^2}{2g}=\frac{V^2-V'^2}{2g}.$$

The remaining energy is

$$\frac{V^2}{2g}-\frac{V^2-V'^2}{2g}=\frac{V'^2}{2g}.$$

A line representing $V'$ may be drawn in the velocity diagram at the right of Plate X, and this combined with $u$ gives $v'$, the real relative velocity at entrance to the moving blades. The loss in the moving blades is

$$L_B=\frac{v'^2}{2g}-\frac{(kv')^2}{2g}=(1-k^2)\frac{v'^2}{2g},$$

where $k=\sqrt{1-y'}$, $y'$ being the per cent loss of energy occasioned as the steam passes through the moving blades. More properly, $y'$ is the percentage of the available energy which is effective in heating the buckets and other steam-passages, and so not effective, at the point under consideration, for producing velocity of flow. The remaining energy, after deducting both losses, is

$$K'=\frac{V'^2}{2g}-(1-k^2)\frac{v'^2}{2g}-\frac{V_1'^2}{2g}$$
$$=\frac{V'^2-V_1'^2-(1-k^2)v'^2}{2g}.$$

These quantities are to be used in the modified velocity diagram at the right on Plate X, and this may now be drawn according to the following assumptions. Let the loss due to

friction in the nozzles correspond to a value of $y=0.12$; and in the buckets let $y'=0.14$. The fraction by which the entrance velocity is decreased is $f$, and the actual velocity of the steam from the nozzles will be

$$V'=fV=V\sqrt{1-y}.$$

Therefore $f=\sqrt{1-y}$, as before stated; and in the present example the value is $\sqrt{0.88}$, or 0.94, approximately. Then $V'=0.94\times3500=3290$ ft. per second. The resulting relative velocity is $v'=2330$, and this is diminished in the buckets to a value $kv'$, where $k=\sqrt{1-0.14}=0.928$. The value of $v_1'$ is then $0.928\times2330=2160$ ft. per second, and the absolute velocity of exit from the buckets is $V_1'=1570$. The nozzle angle of course remains as it was before, but the angle $\Delta'$ has become slightly greater than the corresponding angle $\Delta$ in the ideal case. The work done, per pound of steam, is

$$K'=\frac{3290^2-1570^2-0.14\times(2330)^2}{64.4}=119,000 \text{ foot-pounds.}$$

The work done in the frictionless turbine was found to be

$$K=\frac{3500^2-1870^2}{64.4}=136,000 \text{ foot-pounds.}$$

The efficiency in this ideal case was

$$\frac{3500^2-1870^2}{3500^2}=0.714.$$

The efficiency after deducting the loss due to friction is

$$\frac{119,000}{136,000}\times0.714=0.624.$$

This figure does not represent the true efficiency, because losses due to windage and to friction of journals and stuffing-boxes

have not been considered. Assuming a loss of 10 per cent due to these causes, the work delivered by the machine is

$$0.9 \times 119{,}000 = 108{,}000 \text{ foot-pounds.}$$

The efficiency is therefore 0.566.

Since one pound of steam, in passing through the turbine, causes 108,000 foot-pounds of work to be delivered to the shaft, the steam consumption of the machine in pounds per delivered horse-power hour is

$$\frac{1{,}980{,}000}{108{,}000} = 18.4.$$

Assuming the revolutions of the wheel per minute to be 15,000, the diameter to give a peripheral velocity of 1200 feet per second is

$$\frac{1200 \times 60}{15{,}000 \times 3.14} = 1.53 \text{ feet, or about } 18\tfrac{1}{2} \text{ inches.}$$

If the wheel were to deliver 100 horse-power, it would use 1840 pounds of steam per hour, or about 0.51 pound per second. The nozzle discussed in the example on page 85 would deliver about half of that amount of steam, but five or six nozzles of smaller diameter and length might better be used than two of those referred to.

The dimensions of the nozzles may be found by the same method as used in the previous nozzle calculations.

**The Two-stage Impulse-turbine, with Several Rows of Buckets in Each Stage.**—Let an impulse-turbine have two stages, each containing one set of nozzles, and three rotating and two stationary sets of buckets, as shown in Fig. 60. Let the initial pressure at the throttle-valve be 160 pounds per square inch absolute.

Let expansion in the first nozzles be from 160 pds. to 14 pds. absolute.

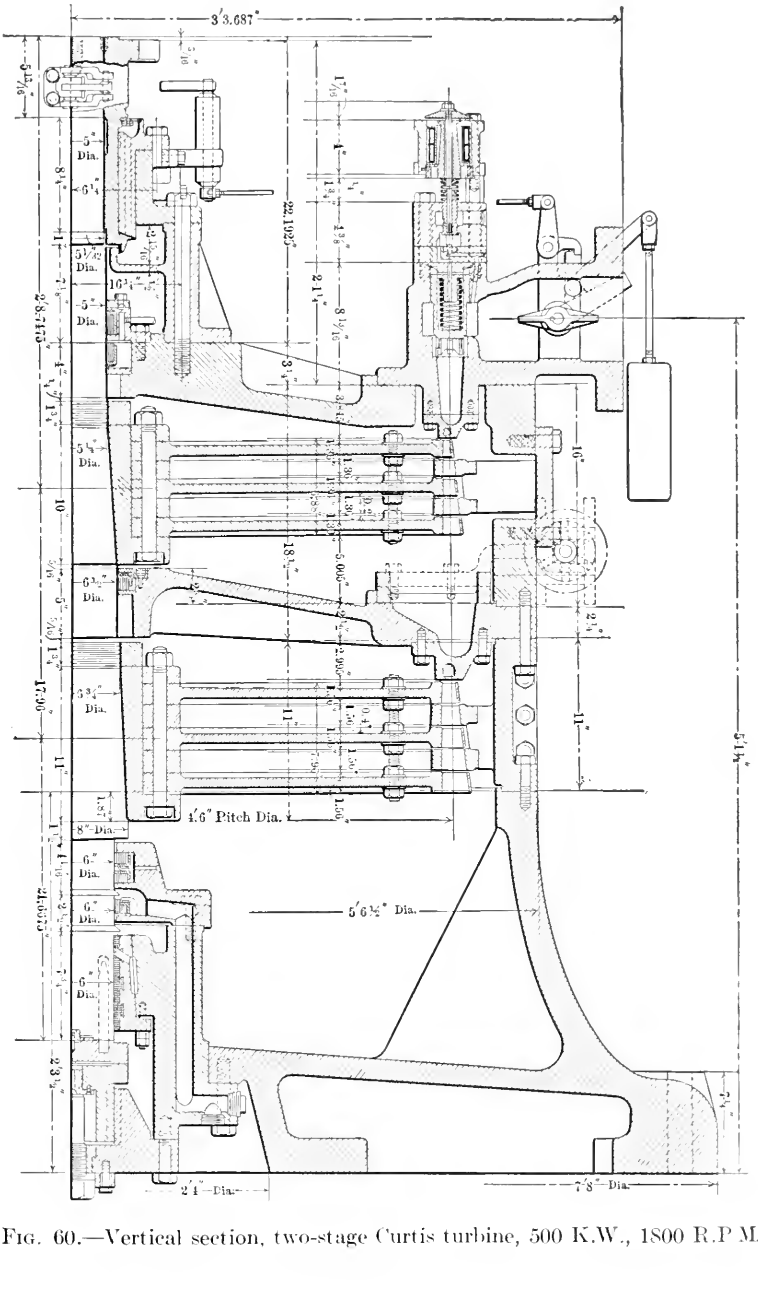


FIG. 60.—Vertical section, two-stage Curtis turbine, 500 K.W., 1800 R.P.M.

Let expansion in the second nozzles be from 14 pds. to a vacuum of 29 inches of mercury.

The ideal case will be considered first, allowing for no losses excepting that due to the energy in the exhaust-steam.

From the curves on Plate XI it is found that the steam during its expansion in the first-stage nozzles gains a velocity of 2990 feet per second. This may be found with the aid of the heat diagram at the back of the book. Thus,

$T_1$, corresponding to 160 pds. absolute, $=824°$ F. abs.
$T_2$, " " 14 " " $=670°$ F. "

Assuming 100% dry steam,—from the chart, the total heat is 1192 B.T.U. per pound at the initial pressure. After adiabatic expansion the heat in the mixture is 1014 B.T.U

$$1192-1014=178 \text{ B.T.U. given up.}$$

The velocity $=V=224\sqrt{178}=2990$ ft. per second, approximately.

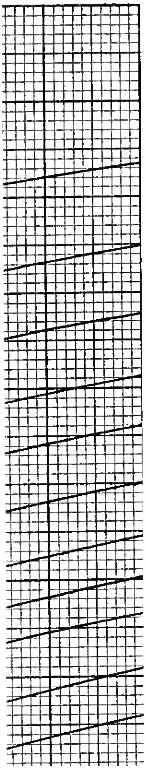
Let the peripheral velocity $u$ be 400 feet per second. This gives a ratio of $\dfrac{u}{V}=0.135$.

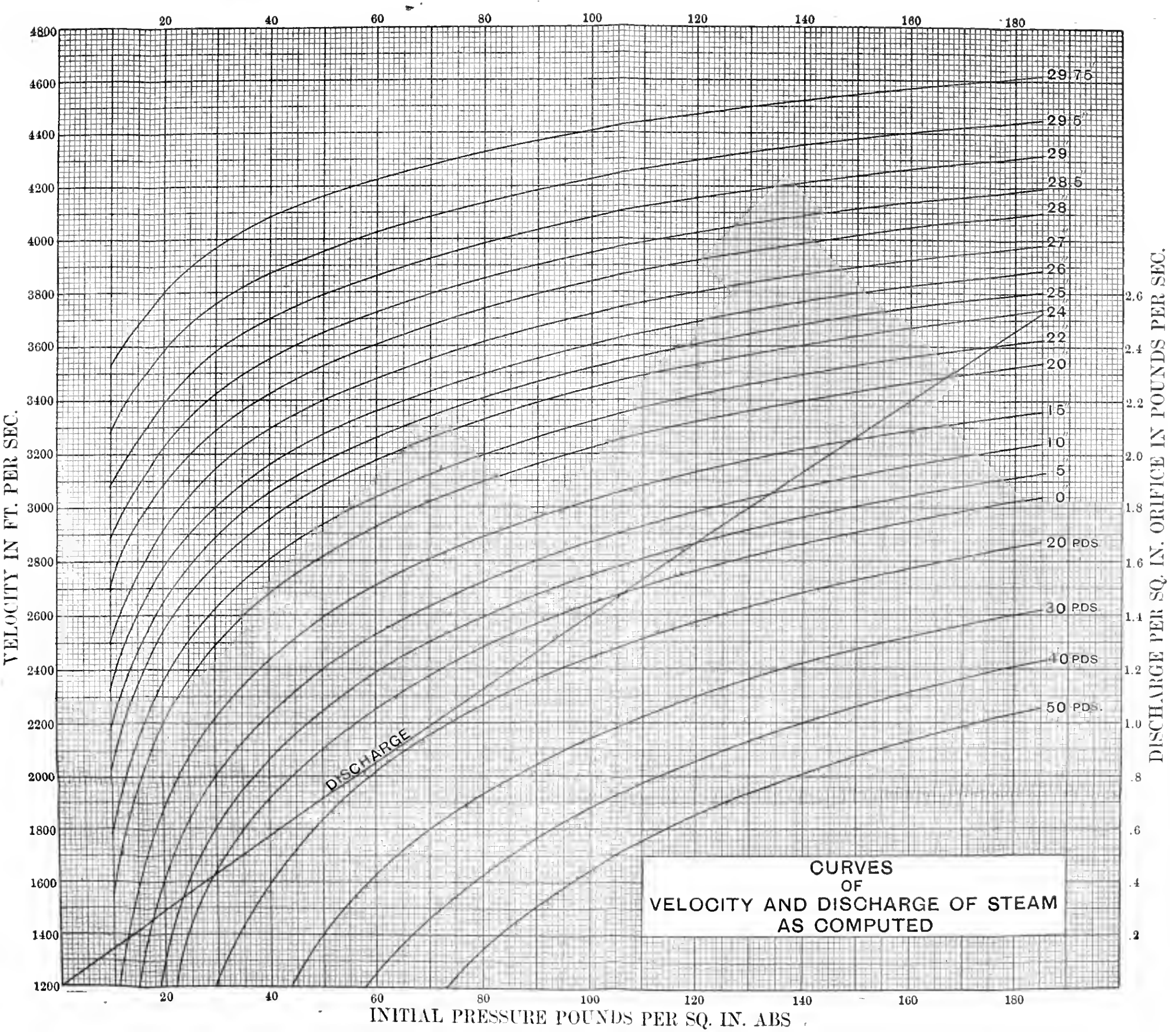
Let the angle of the nozzles with the plane of rotation of the buckets be 20°.

The velocity diagram for the first movable buckets may be drawn as before, the entrance and exit angles of the buckets being the same as those made by the relative velocity lines with the direction of motion of the buckets.

From the relative exit velocity $v_1$ $(=v)$ may be found the absolute velocity $V_1$, and, since the stationary buckets receive the jet in the direction corresponding to the absolute velocity, they may be sketched in, as at *B*. These stationary buckets act as nozzles for the succeeding movable buckets, and the direction of the relative velocity line, $v_2$, is used for determining the angles of entrance and exit for the movable buckets at *C*. In similar manner each stationary and movable set may be outlined.

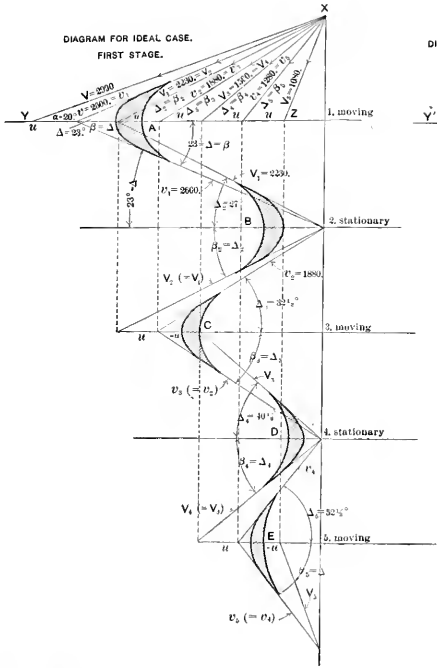
12

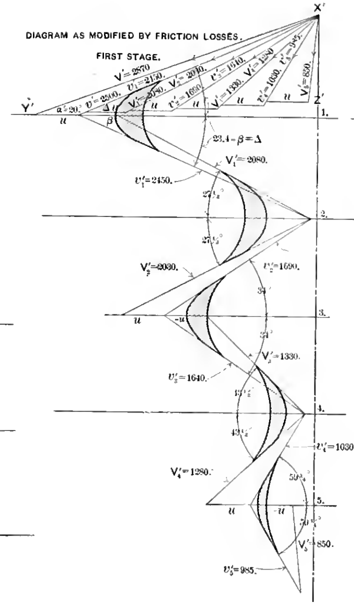


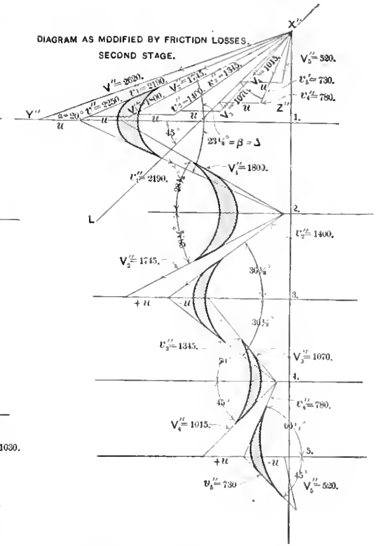



NOTE: FIGURES AT ENDS OF CURVES DENOTE FINAL PRESSURES IN INCHES MERCURY OR POUNDS ABS.

[To face p. 160.




TWO STAGE IMPULSE TURBINE.

The entrance and exit angles of the buckets, whether moving or stationary, are not necessarily made equal to each other, but are modified to suit the energy distribution aimed at in any given case.

[*To face p. 161.*

The efficiency of the system is

$$\frac{V^2 - V_n^2}{V^2},$$

where $V_n$ is the final absolute exit velocity. In the present case there are five sets of buckets, including movable and stationary, and hence $n=5$.

The distance $YZ$, Plate XII, equals $u(n+1)$, and

$$V_n^2 = V^2 + \{(n+1)u\}^2 - 2V(n+1)u \cos \alpha.$$

For the ideal case under consideration the efficiency is

$$\frac{V^2 - V_n^2}{V^2} = \frac{2(n+1)u \cos \alpha}{V} - \frac{\{(n+1)u\}^2}{V^2}.$$

In the single-stage turbine $n=1$ and the efficiency is

$$\frac{4u}{V}\left(\cos \alpha - \frac{u}{V}\right),$$

as was shown on page 23, Chapter I.

In the present case $n=5$; $\cos 20° = 0.94$, approx.

$$\text{Efficiency} = \frac{2 \times 6 \times 400 \times 0.94}{2990} - \frac{(6)^2 \times (400)^2}{(2990)^2} = 0.86+.$$

From the diagram, Plate XII, $V_5 = 1080$ ft. per sec.

$$\text{Efficiency} = \frac{(2990)^2 - (1080)^2}{(2990)^2} = 0.86+.$$

The variation of efficiency with $\alpha$ and with $n$ and $u$ is shown on plate XVII.

The velocity diagram shown at the left on Plate XII is for the ideal case. The velocities represented by the various lines are as follows:

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| $V$ = absolute velocity leaving nozzles. | | | | | | | | |
| $V_1$ = | " | " | " | buckets No. 1. | | | | |
| $V_2$ = | " | " | " | " | " | 2 and equals $V_1$. | | |
| $V_3$ = | " | " | " | " | " | 3. | | |
| $V_4$ = | " | " | " | " | " | 4 | " " | $V_3$. |
| $V_5$ = | " | " | " | " | " | 5. | | |

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| $v$ = relative velocity leaving nozzles. | | | | | | | | |
| $v_1$ = | " | " | " | buckets No. 1 and equals $v$. | | | | |
| $v_2$ = | " | " | " | " | " | 2. | | |
| $v_3$ = | " | " | " | " | " | 3 | " " | $v_2$. |
| $v_4$ = | " | " | " | " | " | 4. | | |
| $v_5$ = | " | " | " | " | " | 5 | " " | $v_4$. |

Since there are no losses during the passage of the steam through the nozzles and buckets, all the energy given up is effective in producing rotation, and the work done may be calculated as follows:

| | | |
|---|---|---|
| In first movable buckets, | $(2990)^2-(2230)^2 \div 64.4 = 61{,}700$ | ft.-lbs. |
| " second " " | $(2230)^2-(1560)^2 \div 64.4 = 39{,}500$ | " |
| " third " " | $(1560)^2-(1080)^2 \div 64.4 = 19{,}600$ | " |
| Total. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . | 120,800 | ft.-lbs. |

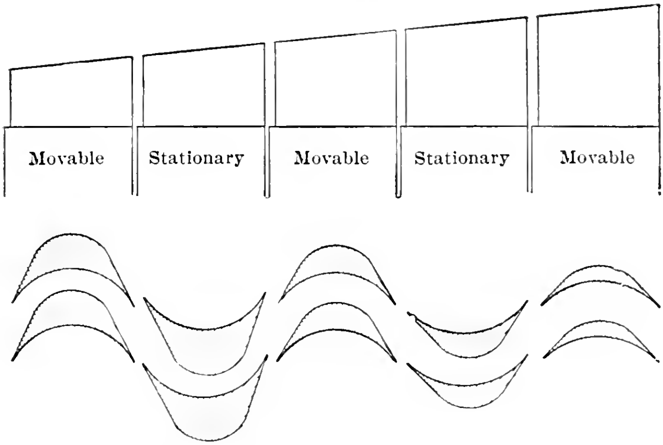
This is to be compared with $V^2 - V_5{}^2 \div 2g$

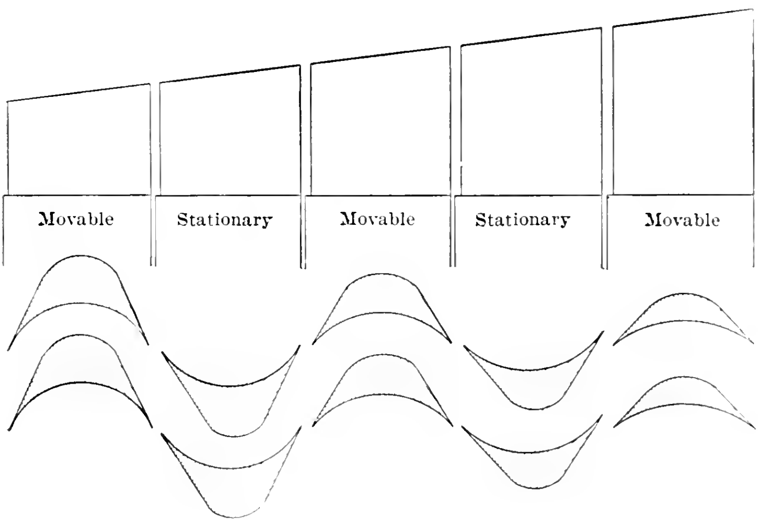
$$= \frac{(2990)^2-(1080)^2}{64.4} = 120{,}800.$$

The efficiency is $\dfrac{(2990)^2-(1080)^2}{(2990)^2} = 0.87.$

The velocities obtained in actual turbines are less than those just considered, because of the frictional resistances encountered by the steam in its passage through nozzles and buckets. The diagram is therefore to be modified accord-

First stage.




Second stage.

Curtis turbine buckets.

ing to the reduction in velocity, and the blade angles made to correspond.

Calling the loss of energy $y$, as before, let the initial velocity $V$ correspond to the value $y=0.08$.

The steam, as it issues from the nozzle, will then have a velocity of

$$V'=224\sqrt{178\times 0.92}=2870 \text{ feet per second.}$$

Let the steam, as it passes through the buckets, fail to gain the full velocity of the ideal case because of frictional resistances represented by the following values of $y$:

| | |
|---|---|
| During passage through set No. 1...... | $y=0.03$ |
| " " " " " 2...... | $y=0.05$ |
| " " " " " 3...... | $y=0.06$ |
| " " " " " 4...... | $y=0.07$ |
| " " " " " 5...... | $y=0.07$ |

The velocities to be used in laying down the diagram will then be:

$V'=2870$ feet per second, as already found.

| | |
|---|---|
| $v_1'=2500\sqrt{1-0.03}$ | $=2450$ feet per second. |
| $V_2'=2080\sqrt{1-0.05}$ | $=2030$ " " " |
| $v_3'=1690\sqrt{1-0.06}$ | $=1640$ " " " |
| $V_4'=1320\sqrt{1-0.07}$ | $=1280$ " " " |
| $v_5'=1020\sqrt{1-1.07}$ | $=985$ " " " |
| $V_5'=$ final absolute velocity | $=850$ " " " |

The resulting modified velocity diagram is shown in the center of Plate XII. The efficiency of this stage of the turbine is not represented, as before, by the difference of the squares of the two absolute velocities,—initial and final, respectively,—for the decrease of the final velocity $V_5$ below the value in the ideal case is due to the fact that the steam is carrying away with it heat energy, which in the ideal case

would be given up as kinetic energy corresponding to increased velocity. The heat carried away is available for doing work in the second stage of the turbine.

The work done on each of the movable sets of buckets may be determined as was done in the case of the single-stage turbine discussed on page 157. Thus, for the nozzles and first moving buckets,

$$V' = fV, \quad \text{where} \quad f = \sqrt{1-y} = \sqrt{1-0.08} = 0.96.$$

Therefore $\quad V' = 0.96 \times 2990 = 2870$ feet per second.

The work done on the first moving buckets is

$$K_1' = \{(V')^2 - (V_1')^2 - (1-k_1^2)v'^2\} \div 2g$$
$$= \frac{2870^2 - 2080^2 - 0.03 \times 2500^2}{64.4} = 58{,}000 \text{ ft.-pds.}$$

Similarly, the work done on the second moving buckets, that is, on set No. 3, is

$$K_3' = \{(V_2')^2 - (V_3')^2 - (1-k_3^2)v_2'^2\} \div 2g$$
$$= \frac{2030^2 - 1330^2 - 0.06 \times 1690^2}{64.4} = 33{,}800 \text{ ft.-pds.}$$

Finally, the work done on the last moving buckets (set No. 5) is

$$K_5' = \{(V_4')^2 - (V_5')^2 - 0.07 \times (v_4')^2\} \div 2g$$
$$= \frac{1280^2 - 850^2 - 0.07 \times 1030^2}{64.4} = 13{,}100 \text{ ft.-pds.}$$

The total work done on the wheels by the steam, per pound, is the sum of these amounts, or 104,900 foot-pounds.

In the ideal case the work was 120,800 foot-pounds, and the efficiency was 0.87.

The efficiency in the present case is

$$\frac{104,900}{120,800} \times 0.87 = 0.755.$$

The steam consumption of this turbine, if no further stage were added, would be

$$\frac{1,980,000}{104,900} = 18.9 \text{ pounds per horse-power hour.}$$

If there were a loss of 10%, due to friction of journals and to windage, as was assumed in the case of the simple impulse-turbine of one rotating wheel, the steam consumption of the first stage of turbine in the present example, if worked alone, would be about 21 pounds per horse-power hour. This is about 12% higher than that of the simple turbine, but the important difference between the two machines lies in the fact that, while the simple turbine considered has a peripheral speed of 1200 feet per second, and a ratio of initial steam velocity to peripheral velocity of 2.9 to 1, the turbine with three rotating wheels develops power with about equal economy when working at a peripheral velocity of 400 feet per second, or one third that of the simple turbine, and with a ratio of peripheral to initial steam velocity of about 1 to 7.2. It is to be remembered, also, that the simple turbine considered is assumed to exhaust into a condenser, although it has somewhat low nozzle efficiency; while the turbine with three rotating wheels is assumed to be exhausting at about atmospheric pressure. This was done in the present example in order that the effect of adding a second set of nozzles and three more rotating wheels might be shown, and it remains to investigate that part of the problem.

**Calculations for the Second Stage of the Turbine.**—From the heat diagram it was found, in the first part of the example, that steam in expanding adiabatically from 160 to 14 pounds absolute pressure gave up 178 B.T.U. per pound. In a fric-

tionless and otherwise ideal turbine all of this energy would be effective in producing velocity of flow in the nozzles. The frictional resistance opposed by the surfaces of nozzles and buckets causes the steam to give up less heat as work on the buckets, and therefore to carry away more heat into the exhaust, than it would in a frictionless turbine. The useful work done upon the buckets of the three moving wheels considered has been found to be 104,900 foot-pounds per pound of steam. This is equivalent to 135 B.T.U.

If losses caused by leakage past the buckets, and by mechanical friction, windage, etc., be neglected, the steam at exhaust from the last of the three movable buckets will possess an amount of heat greater than it would have possessed after purely adiabatic expansion, equal to $178-135=43$ B.T.U. per pound. After adiabatic expansion, if such had occurred, from 160 to 14 pounds absolute, the steam would contain 1018 B.T.U. per pound, and its quality would be 0.868. The heat of vaporization of dry saturated steam at 14 pounds absolute is 967 B.T.U. There is present in each pound of the mixture of steam and water $1.00-0.868=0.132$ pound of water, and to evaporate this would require $0.132\times 967 = 128$ B.T.U. The amount of heat available for accomplishing evaporation, and therefore for increasing the quality of the steam, is 43 B.T.U. This is sufficient to increase the quality by $\frac{43}{128}\times 0.132=0.0443$. The quality of the steam entering the second-stage nozzles will then be $0.868+0.044=0.912$.

Steam of 14 pounds absolute pressure and 0.912 quality contains 1060 B.T.U. per pound. This steam is to expand in the second-stage nozzles to a final pressure corresponding to a vacuum of 29 inches of mercury or a temperature of 540 degrees absolute. Following the vertical line on the heat diagram from the state-point for the steam before it enters the second-stage nozzles down to the line of 540 degrees absolute temperature, the heat contents of the mixture of steam

and water, after expansion, is 875 B.T.U. The heat available for producing velocity in the jet from the second-stage nozzles is then $1060-875=185$ B.T.U.

The heat employed in the first stage was.. 135 B.T.U.

Total .................... 320 B.T.U.

The total heat drop during expansion of dry saturated steam from 160 pounds absolute to a vacuum of 29 inches is 320 B.T.U., in case the quality of the exhaust is as indicated by the above calculation, that is, 0.78. The quality after adiabatic expansion would of course be lower than this. Let the energy loss due to friction in the second-stage nozzles be that corresponding to a value of $y=0.26$. The initial velocity of steam, as it strikes the first buckets, will then be

$$V''=224\sqrt{185\times0.74}=2620 \text{ feet per second.}$$

Let the values of $y$ for the second stage be as follows:

During passage through set No. 1......$y=0.05$
" " " " " 2......$y=0.06$
" " " " " 3......$y=0.08$
" " " " " 4......$y=0.10$
" " " " " 5......$y=0.12$

The velocities will then be as follows:

$V''=2620$ feet per second, as already found.
$v_1''=2250\sqrt{1-0.05}$ $=2190$ feet per second.
$V_2''=1800\sqrt{1-0.06}$ $=1745$ " " "
$v_3''=1400\sqrt{1-0.08}$ $=1345$ " " "
$V_4''=1070\sqrt{1-0.10}$ $=1015$ " " "
$v_5''=780\sqrt{1-12}$ $=730$ " " "
$V_5''=$ final absolute velocity $=520$ " " "

As the losses increase, the blade angles become greater and greater, and the designer may decide to limit the size of exit angle. Suppose, for example, it were thought advisable to limit the exit angles to 45° or less. The angle of $V_4''$ would become larger than 45° if the method of laying out the diagram were not changed. A line $X''L$ may be drawn making an angle of 45° with the line of action of the buckets, and $v_4''$ may be revolved so as to coincide with $X''L$. Completing the diagram as shown, by measuring off each succeeding velocity line, as $v_5''$ upon $X''L$, the corresponding velocities may be found, and the exit angles of the buckets made as desired. A similar change might have been made in the diagram for the first stage, and would have resulted in smaller exit angles for the last buckets. This would have slightly increased the efficiency of the first stage, but that it would have improved the turbine as a whole is doubtful.

The work done by the steam upon the moving buckets of the second stage may be calculated as was done for the first stage.

For the first moving buckets,

$$K_1'' = \frac{(2620)^2 - (1800)^2 - 0.05 \times (2250)^2}{64.4} = 52{,}200 \text{ ft.-pds.}$$

$$K_3'' = \frac{(1745)^2 - (1070)^2 - 0.08 \times (1400)^2}{64.4} = 27{,}000 \quad \text{``}$$

$$K_5'' = \frac{(1015)^2 - (520)^2 - 0.12 \times (780)^2}{64.4} = 10{,}700 \quad \text{``}$$

| | |
|---|---|
| Total work of second stage . . . . . . . . | 89,900 ft.-pds. |
| Work of first stage of turbine, 104,900, say | 105,000 |
| Total work of turbine, per pound of steam, | 194,900 |

Taking the losses due to friction of journals, windage, and leakage as 22 per cent of the work done by the steam, the steam consumption of the turbine is $\dfrac{1{,}980{,}000}{195{,}000 \times 0.78} = 13$ pounds

per delivered horse-power hour, approximately, or 17.4 pounds per K.W. hour.

These calculations are based upon saturated steam at the throttle-valve. When superheated steam is used the losses are much lower and the economy correspondingly higher. This is shown in the tables of performance of the various turbines, the steam consumption being as low as 11.3 pounds per electrical horse-power hour when operating with 200 degrees F. superheat. This means 15.1 pounds per K.W. hour.

Up to this point nothing has been said as to the amount of power the turbine is to develop. It has been shown that the steam consumption per delivered horse-power at the turbine shaft may be expected to be 13 pounds. This economy refers to the full-load conditions, and the steam consumption will increase at loads below and above full loads. If the turbine is intended for operating an electric generator having an efficiency of 0.88, the steam used per electrical horse-power hour will be 14.8 pounds at full load. This will be increased by from 15% to 20% at 50% overload. Taking the increase as 15%, the steam consumption at 50% overload will be about 17.4 pounds per electrical horse-power hour.

Let the turbine be required to operate a generator delivering 400 electrical horse-power at full load, and 600 electrical horse-power when called upon for maximum overload. The total amount of steam required will be as follows:

Full load, $W = 14.8 \times 400 \div 3600 = 1.65$ pounds per second.

At 50% overload, $W' = 17.4 \times 600 \div 3600 = 2.9$ pounds per second.

*To find the diameter of the turbine wheels, and the area for passage of steam through the second-stage nozzles.*—The peripheral velocity of buckets having been decided upon during the design of the buckets, the rate of revolution of the turbine fixes the diameter of the wheels. Let the R.P.M. be 2000. Then for a peripheral velocity of 400 ft. per second the mean diameter of bucket circle will be

$$\frac{400\times 60}{3.14\times 2000}=3.82 \text{ feet or } 46 \text{ inches.}$$

It has been assumed in the present problem that the steam pressure at entrance to the second-stage nozzles will be 14 pounds absolute. The second-stage nozzles will be non-expanding, while those of the first stage will be expanding nozzles. The pressure in the throat of the second-stage nozzles will be about $.577\times 14=8.10$ pounds absolute. The total loss of energy in these straight nozzles has been assumed to be that corresponding to $y=0.26$ (see page 168). Assuming that the steam expands to the shell pressure before entering the first row of buckets in the second stage, the initial velocity has been shown to be 2620 feet per second (see page 168). But this is not the velocity in the entrance or orifice of the nozzles. Let the friction loss in the orifice be represented by $y=0.08$. The heat contents at entrance to the nozzles is 1060 B.T.U. per pound. The steam is to fall in pressure at once upon entering the nozzles, to 8.1 pounds, and to be dried to a certain extent during this drop in pressure. The initial quality is 0.912 (page 167), and if the expansion to 8.1 pounds shoud be adiabatic the heat diagram shows that the quality in the orifice would be $x'=0.885$ and the heat contents 1023 B.T.U. per pound. Since the heat of vaporization at 8.1 pounds absolute is 986, the increase in quantity due to the heat of friction will be $x''=.08(1060-1023)\div 986.=0.003$ (see page 83). The quality in the orifice will then be

$$x'+x''=0.885+0.003=0.888.$$

The corresponding heat contents is 1028 B.T.U. per pound. The velocity in the orifice is $224\sqrt{(1060-1028)}=1255$ feet per second. Since the specific volume of dry steam at 8.1 pounds absolute is 47 cubic feet per pound, that at 0.888 quality will be $47.0\times 0.888=41.7$ cubic feet. The necessary cross-sectional area of the orifices, collectively, will then be

$$A=\frac{2.9\times 41.7\times 144}{1255}=13.9 \text{ square inches.}$$

The nozzles through which the steam expands into each shell of the turbine are ordinarily of four-sided cross-section, slightly rounding at the throat in some cases; but each nozzle presents a four-sided outlet (*ABCD*, Fig. 64), next to the first row of buckets. The radial walls are often formed of steel

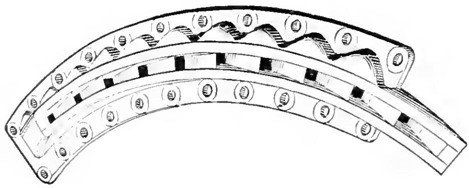


FIG. 63.

plate about $\frac{1}{16}$ inch thick, cast into the nozzle frame as shown in Fig. 63.

The general arrangement of turbine casing and nozzles is shown diagrammatically in Fig. 64, the pitch of nozzles being greatly exaggerated in this diagram. The steam is led into the first stage of the turbine through expanding nozzles, and into the succeeding stage or stages through straight nozzles, of uniform cross-sectional area. These nozzles are short, and the exit ends are cut off parallel to the plane of wheel rotation.

The first-stage nozzles may occupy only a small part of the annular space available for them; but in the final stage, owing to the great volume of steam to be passed, it may be necessary to utilize the entire available space. If the turbine should be small in diameter, comparatively, the nozzles might require to be of such height radially that the buckets, especially the last row of the stage, would be higher than good practice permits. The whole circumference is not ordinarily available for nozzles, because of structural conditions; for example, the diaphragms may be made in halves, and the flanges for joining the two parts take up some space. In any case, there is a certain angle at the center of the shaft, which can conveniently be subtended by the nozzles. The latter may be disposed in two groups, each subtending half the total angle available, one group being

in each half of the diaphragm. It becomes necessary to determine the height $H$ which the nozzles must have, to afford the requisite cross-sectional area of steam passage.

Referring to Fig. 64, the angle $\Delta$ which the design permits the nozzles to subtend, and the other particulars to which

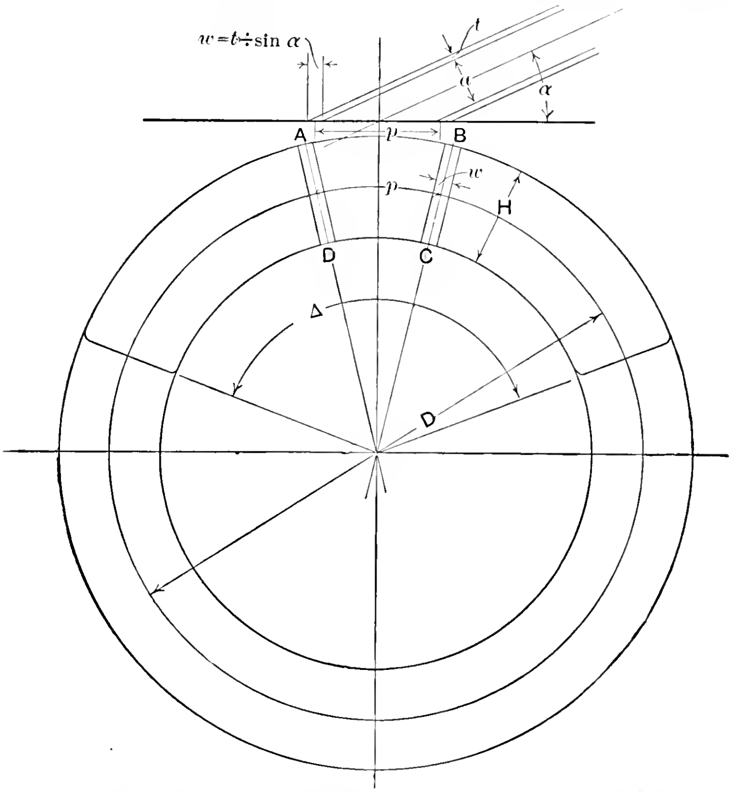



FIG. 64.—Diagrammatic representation of nozzles in a Curtis turbine. The pitch of nozzle walls is purposely exaggerated.

symbols have been given, are related to each other in the following manner. The fraction of the pitch of nozzles, $p$, which represents clear opening in an axial direction (axial with respect to the turbine axis) is

$$k=\frac{p-w}{p}.$$

But $$w=\frac{t}{\sin\alpha} \text{ and therefore } k=1-\frac{t}{p\sin\alpha}.$$

Supposing, for example, a turbine having a pitch diameter of 46 inches, as in the present example, should have nozzle walls made of $\frac{1}{16}$-inch plate, and that the angle $\alpha=20°$ for the nozzles of the last stage. Let the pitch, $p$, $=1.46$ inches.

Then, $$k=1.-\frac{0.0625}{1.46\times0.342}=0.875.$$

This means that the nozzle walls occupy $12\frac{1}{2}\%$ of the space devoted to nozzle openings in front of the buckets.

The nozzles, as a whole, subtend an angle of $\Delta$ degrees at the center of the diaphragm, and the whole length of arc of pitch-circle included by the angle $\Delta$ is $\frac{\pi D\Delta}{360}$ inches. Then the mean net length of the space occupied by nozzle outlets, after taking out the area occupied by the ends of the nozzle walls, is equal to

$$\frac{k\pi D\Delta}{360}.$$

The area perpendicular to the direction of steam flow through the nozzles is, then,

$$A=\frac{k\pi D\Delta H\sin\alpha}{360}, =0.0087HDk\Delta\sin\alpha.$$

Applying this to the case in hand, the required area $A$ is 13.9 square inches; $D=46$ inches; $\alpha=20°$; $\sin\alpha=0.342$. Let the angle $\Delta=120°$. The necessary height of nozzles will then be

$$H=\frac{13.9}{0.0087\times46\times0.875\times120\times0.342}=0.97 \text{ inches.}$$

The height of the buckets nearest the outlet end of the nozzles is made about $2\frac{1}{2}\%$ greater than the nozzle height. This would make the first row of buckets in the present case

0.995, or approximately 1 inch high at the steam-inlet side. The ratio between the maximum height of the last row of buckets in a given stage and the minimum height of the first row, is called the "height-ratio." If this should be made equal to 2, in the present case, the maximum height of the last bucket would be 2 inches.

The meaning of the term "height-ratio" will be understood by reference to the figures on page 163. The relative areas for passage of steam through the successive rows of buckets are of more value than are the height-ratios; but the latter, with given bucket-shapes and spacing, serve as something of an indication of the value of area ratios.

*Efficiency of steam turbines. Design of impulse-turbines on the basis of experimentally determined stage efficiency. Heat analysis of steam turbines.*—Steam-turbine efficiency is ordinarily expressed as the ratio of the work actually delivered from the turbine shaft per unit of time, to that which would have been delivered if the steam had expanded adiabatically, and the total energy available from such expansion had been transformed into mechanical work. As an example, suppose the initial steam pressure to be 165 pounds absolute per square inch, and that the steam were superheated 100 degrees F. From the chart at the back of the book the steam would contain, in its initial condition, 1252 B.T.U. per pound. If the steam should expand adiabatically to a pressure of 1 pound absolute (562 degrees), its final heat contents, found by passing down an adiabatic line on the chart to 562 degrees, would be 910 B.T.U. A perfect engine would deliver mechanical energy equivalent to the difference in heat contents between the initial and final states of the steam, and would completely utilize, therefore, $1252-910=342$ B.T.U. per pound of steam used. This is called the *available heat*, H, per pound of steam, and is equivalent to 778 H, foot-pounds, or, in this case,

$$778 \times 342 = 266{,}076 \text{ foot-pounds.}$$

Since the expression "one horse-power" means an expenditure of 33,000 foot-pounds per minute, or 1,980,000 foot-pounds

per hour, the number of pounds of steam which a perfect engine would require per horse-power hour under the above conditions, is

$$\frac{1,980,000}{266,076}=7.4 \text{ approximately.}$$

If an actual engine, operating under the same conditions, uses 13 pounds of steam per delivered horse-power hour, its efficiency is $7.4 \div 13 = 0.57$.

The efficiency of any steam engine may be calculated in a similar manner, from results of tests; thus

$$\text{Efficiency} = \frac{1,980,000}{\text{water rate} \times \text{available energy in foot pounds per pound of steam per hour.}}$$

The table given below shows the use which may be made of such calculations in determining the effect upon efficiency produced by varying conditions of operation.

Calculating from the water rates given below the variation of efficiency with load and with superheat is shown in the following table. (Particulars of turbine given below.)

| 100° Superheat. | | | | Saturated Steam. | | | |
|---|---|---|---|---|---|---|---|
| Test No. | B.H.P. | W.R. | Effic. $=7.8 \div$ W.R. | Test No. | B.H.P. | W.R. | Effic. $=8.35 \div$ W.R. |
| 1 | 269. | 16.2 | .481 | 1 | 245. | 19.4 | .430 |
| 2 | 402. | 14.6 | .534 | 2 | 406. | 16.2 | .516 |
| 3 | 649. | 13.3 | .586 | 3 | 650. | 14.6 | .572 |
| 4 | 766. | 13.1 | .595 | 4 | 716. | 14.7 | .568 |
| 5 | 956. | 13.5 | .577 | 5 | 1144. | 15.7 | .532 |
| 6 | 1195. | 14.1 | .553 | | | | |

Westinghouse-Parsons 400 K.W. Steam Turbine, 3600 R.P.M., with automatic by-pass valve. Work absorbed by water-brake. Tests to determine economy to be gained by use of 100° F. superheat. Steam-pressure in main steam-pipe 150 pounds gage or 165 pounds absolute in both cases below. Vacuum 27 inches in both cases. Bucket speed varying from

157 feet per second at H.P. end to 345 feet per second at L.P. end. Available energy at 100° superheat, assuming adiabatic

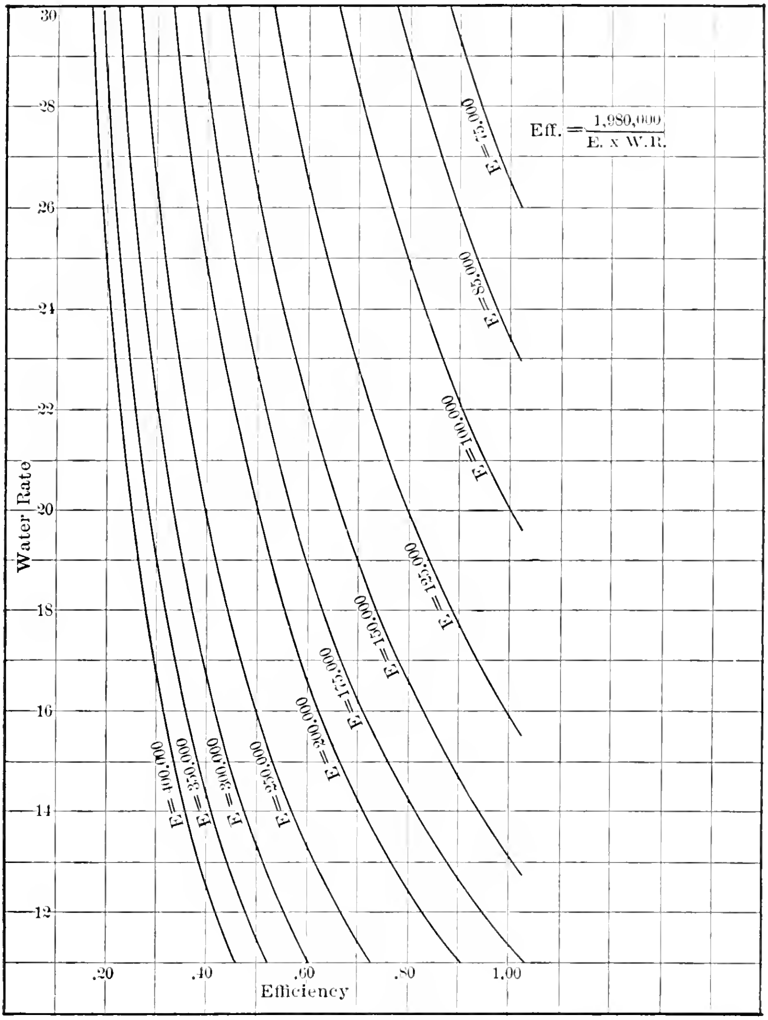



FIG. 65.—Curves of Efficiency and Water-rate for given Available Energy.

expansion to 27 inches vacuum $=326$ B.T.U. or 254,000 foot-pounds per pound of steam.

$$\text{Efficiency} = \frac{1{,}980{,}000}{\text{W.R.} \times 254{,}000} = 7.8 \div \text{W. R.} \quad \text{where} \quad \text{W. R.} =$$

pounds steam per B.H.P. hour. With saturated steam, and

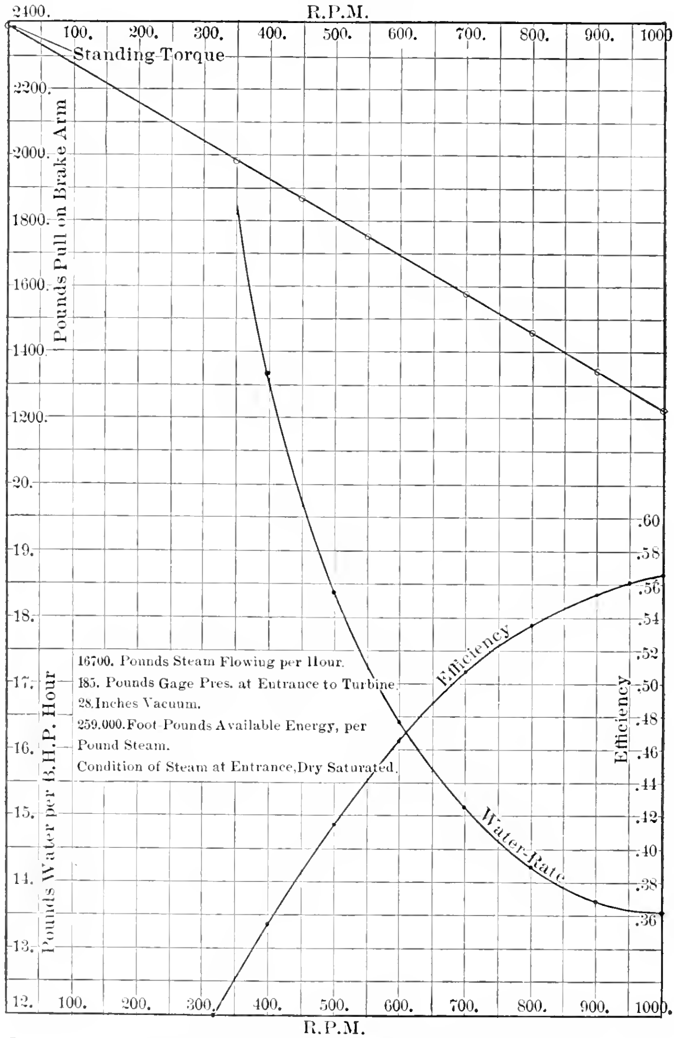



FIG. 66.—Curves of Efficiency and Water-rate as Computed from Experimentally Determined Torque Line.

same vacuum, available energy = 237,000 foot-pounds, and

$$\text{efficiency} = \frac{1{,}980{,}000}{\text{W.R.} \times 237{,}000} = 8.35 \div \text{W.R.}$$

The efficiency corresponding to a given amount of available energy, and given water-rate, may be found approximately from the curves in Fig. 65, without calculation.

If a brake has been used to absorb and measure the work done by a turbine, a "torque-line" may be plotted from the results of the test, as shown in Fig. 66. With a given constant rate of steam flow the pull on the brake-arm varies inversely as the speed of revolution of the turbine. If the shaft be brought to rest by the brake and the steam caused to pass through the turbine as before, the pull on the brake is greater than when the shaft is in motion. This is called the "standing-torque." If the shaft is permitted to rotate, the torque decreases uniformly with increase of speed of rotation. The torque-line is straight in all cases, as shown in Fig. 66. From the torque-line and other results of tests, curves of water-rate and efficiency may be plotted, based upon such calculations as are outlined below.

Let $P.$ = pull on brake-arm, pounds.
$B.H.P.$ = brake horse power.
$W$ = pounds of steam per hour, total.
$W.R.$ = water-rate, or pounds of steam per B.H.P.-hour.
$E.$ = available energy, foot-pounds per pound of steam used.
$Eff.$ = effiiciency of turbine, or of the part tested.
$r.$ = length of brake-arm, feet.
$R.P.M.$ = revolutions per minute.

Then,
$$B.H.P. = \frac{2\pi rP \times (\text{R.P.M.})}{33{,}000},$$

$$W.R. = \frac{33{,}000\ W}{2\pi rP \times (\text{R.P.M.})}.$$

But
$$W.R.\ \text{also} = \frac{1{,}980{,}000}{Eff. \times E}.$$

Therefore $$\frac{1{,}980{,}000}{Eff.\times E}=\frac{33{,}000\,W}{2\pi rP\times(\text{R.P.M.})},$$

and $$Eff.=\frac{1{,}980{,}000\times 2\pi rP\times(\text{R.P.M.})}{33{,}000\,W\times E}=\frac{377rP\times(\text{R.P.M.})}{W\times E}.$$

To illustrate the use of this expression for efficiency, supposing a torque-line such as is shown in Fig. 66, has been obtained, and curves of efficiency and water-rate are to be plotted from it. Let $r=5.25$ feet; $W=16{,}700$ pounds per hour; $E=259{,}000$ foot-pounds available per pound of steam. The revolutions per minute and the corresponding values of $P$ are taken from the torque-line.

| R.P.M. | $P$. | $\text{Eff.}=\frac{377rP\times(\text{R.P.M.})}{W\times E}$. | $\text{W.R.}=\frac{1{,}980{,}000}{\text{Eff.}\times E}$. |
|---|---|---|---|
| 400 | 1925 | .353 | 21.7 |
| 600 | 1690 | .465 | 16.4 |
| 800 | 1460 | .535 | 14.3 |
| 1000 | 1230 | .563 | 13.6 |

The efficiency of complete turbines, and also of component stages if tested by themselves, may be ascertained in the manner indicated. From a knowledge of the efficiency of the component parts of a turbine under working conditions, calculations may be made as to the probable efficiency of proposed combinations of those parts into complete turbines. The Rateau and Curtis types, consisting of a number of separate wheels, each in a separate compartment, are especially well adapted to such analysis. In an experimental turbine, specially arranged for the purpose, each stage, consisting of a set of nozzles and one or more sets of buckets, may be tested by itself. Or certain stages, if not each one by itself, may be taken to represent average conditions, and the efficiency and capacity of the various stages and combinations of stages may be ascertained by a properly arranged series of tests.

In the calculations for efficiency given above, the loss by friction of the shaft in the bearings, etc., is included. This should obviously be allowed for only once in a complete turbine,

and not in connection with each stage tested. The efficiency of each stage may be calculated on the basis of "bucket horse-power," or the power represented by the pull on the buckets, independently of the mechanical friction and the windage losses, these being astertained by separate experiments.

During the development of the turbine, experience accumulates indicating the number of compartments or stages to be given to impulse turbines, and the number of "steps-up" in diameter of turbines of the Parsons type. In general, as higher initial pressures and degrees of superheat are used the number of stages is increased. Such particulars, and those concerning the number of rows of movable and stationary buckets to be used in each stage of impulse turbines in order that certain efficiences may be obtained; the magnitude and variation of bucket angles best suited to the energy distribution aimed at in the various stages; the proportions of nozzles, and the relative heights of nozzles and buckets,—such questions are determined by experiment, calculation, and scientific research of various kinds concerning the action of the steam as it passes through the turbine. With all types of steam-turbine at present under development such work is being done, and refinements in methods of analysis, calculation, and construction are resulting in improvement in economy and in operation.

**As an example** of the way in which steam-turbines may be proportioned upon the basis of such investigations as have been discussed in the preceding paragraphs, let it be decided to design a turbine of the several stage, velocity-compounded type. The efficiency of the different stages at various buckets-speeds may be supposed to have been determined and plotted in the form of curves showing the variation of efficiency with bucket-speed and available energy.

The question of rate of revolution and corresponding bucket-speed determines the diameter of the turbine wheels. The revolutions are decided upon according to the speed at which it is desired to rotate the shaft of, for example, a generator, or propeller wheel to which the turbine is to be connected. The efficiency is directly dependent upon bucket-speed and available

energy. The efficiencies it is necessary to use are those obtained experimentally with buckets and nozzles similar to those to be used in the proposed turbine. It should, therefore, be possible to predict closely what each stage will do in the completed machine. The first stage of a velocity-compounded turbine may be given such bucket angles and such a number of rows of buckets that it will absorb a greater percentage of the available energy than is absorbed by any one of the succeeding stages. The efficiency of the first stage may be somewhat lower than that of the others, but as it is affected by a greater heat drop than is allowed in the other stages, the work done by the first is in general greater than that done by any other stage.

In order to proportion the steam-nozzles leading from each compartment, or shell, to the next, so as to obtain the energy distribution aimed at in the turbine, calculations are made as shown in tabular forms A and B on pages 183–184. The results of these calculations relate to the condition of the steam as to pressure, quality, temperature, etc., at the entrance to the nozzles of each stage. From this information the cross-sectional areas of the nozzles are determined so that the requisite amount of steam may be discharged into the buckets, per unit of time, to give the desired horse-power.

The general scheme of calculation is similar to that used in the example worked out on more completely theoretical lines, on pages 158 to 173, but in the present case there is no attempt to definitely locate and allow individually for the frictional and other losses in each stage. The experimentally determined stage efficiency takes account of all losses excepting windage and shaft friction, which are allowed for separately, and avoids the necessity of detailed analysis.

The principle followed in calculating for the steam condition in the various stages is as follows: Steam possessing a known amount of available energy per pound is supposed to drop in pressure and temperature during its passage through each stage until it gives up a certain predetermined proportion of its available energy. This drop is supposed to take place

FORM A. TURBINE CALCULATIONS.

Initial Steam Pressure, 165. pounds per square inch absolute.
Initial Superheat, 100. degrees Fahr.
Initial Heat Contents at Bowl of First Stage Nozzle, $HB_1$ .................... = 1252. B.T.U.
Final Heat Contents after adiabatic expansion to 28. inches vac., $H_2$ ........... = 910. B.T.U.
Available Energy, assuming adiabatic expansion, $HB_1 - H_2$ ..................... = 312. B.T.U.
Energy distribution aimed at; $0.30(HB_1 - H_2)$ in first stage, $h_{s_1}$ ............... = 102. B.T.U.
Energy distribution aimed at; $0.14(HB_1 - H_2)$ in remaining stages $= h_{s_2}$, $h_{s_3}$, etc. = 48. B.T.U.
Size of Turbine, 2000. horse power; 96 inches, mean diameter.
Revolutions per minute, 1000; mean bucket speed, 420 feet per second.

| Stage Number. | 1. | 2. | 3. | 4. | 5. | 6. |
|---|---|---|---|---|---|---|
| * Heat contents at bowl of nozzle, $HB_1$ etc., B.T.U. | 1252. | 1203. | 1177. | 1151. | 1125. | 1099 |
| Bucket speed, feet per second | 420. | 420. | 420. | 420. | 420. | 420. |
| Stage efficiency, $e_1$, $e_2$, etc. | 0.48 | 0.55 | 0.55 | 0.55 | 0.55 | 0.55 |
| Available energy for the stage, $h_{s_1}$, $h_{s_2}$ etc., B.T.U. | 102. | 48. | 48. | 48. | 48. | 48. |
| Heat present after adiabatic expansion, $HB_1 - h_{s1}$, $HB_2 - h_{s_2}$, etc., B.T.U. | 1150. | 1155 | 1129. | 1103. | 1077. | 1051. |
| Reheat, $h_{r_1}$, $h_{r_2}$, etc., $h_{r_1} = (1.00 - e_1)h_{s_1}$ etc., B.T.U. | 53. | 22 | 22. | 22. | 22. | 22. |
| Pressure in shell, pounds absolute, from chart | 18. | 21.5 | 12.5 | 6.0 | 2 5 | 1.0 |
| Temperature in shell, degrees F., from chart | 339. | 277. | 219. | 170. | 135. | 106. |
| Temperature in shell, degrees F., absolute | 800. | 738. | 680. | 631. | 596. | 567. |
| Quality at exit from shell, per cent, from chart | ........ | ........ | ........ | 99.2 | 97 6 | 96.1 |
| Superheat at exit from shell, degrees F., from chart | 61. | 38. | 15. | ........ | ........ | ........ |

* With exception of the quantity for stage No. 1, this line cannot be filled out until the remaining calculations called for on this sheet have been made.

FORM B.

## CALCULATION OF AREA OF THROAT IN TURBINE NOZZLES.

Horse power of turbine 2000
Water rate, 14.2 pounds per B.H.P hour.
Total Water per second 7.9 pounds $=W$.

| | Stage Number. | | | | | |
|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 |
| Initial absolute pressure at entrance to nozzles, from curve, $P_{B_1}$, $P_{B_2}$, etc | 165. | 48 | 24.5 | 12.5 | 6.0 | 2.5 |
| Degrees F. superheat at nozzle bowls | 100. | 61. | 39. | 16. | ........ | ........ |
| Temperature of saturated steam at bowl pressure, degrees F. | 366. | 278. | 239. | 203. | 169. | 134. |
| Initial temperature at entrance to bowl, degrees F. | 466. | 339. | 278. | 219. | 169. | 134. |
| Orifice pressure $.577P_{B_1}$, $.577P_{B_2}$, etc., pounds absolute | 95. | 27.7 | 14.2 | 7.2 | 3.46 | 1.44 |
| Heat contents in bowl, $H_{B_1}$, $H_{B_2}$, etc., B.T.U. | 1252. | 1203. | 1177. | 1151. | 1125. | 1099. |
| Heat contents in orifice, assuming ad. exp. (from chart), $H_{O_1}$, $H_{O_2}$, etc., B.T.U. | 1208. | 1160. | 1150. | 1120. | 1095. | 1070. |
| Heat given up in orifice, $H_{B_1}-H_{O_1}$, $H_{B_2}-H_{O_2}$, etc., BT.U. | 44. | 43. | 27. | 31. | 30. | 29. |
| Efficiency of orifice, $k_1$, $k_2$, etc | 0.96 | 0.95 | 0.92 | 0.92 | 0.90 | 0.90 |
| Velocity of flow in orifice $224\sqrt{k_1(H_{B_1}-H_{O_1})}$, etc., $=V_O$ ft/sec | 1460. | 1430. | 1120. | 1195. | 1165. | 1145. |
| Superheat in orifice, from chart, degrees F. | 43. | 5.0 | 2.0 | ........ | ........ | ........ |
| Quality in orifice, from chart, per cent. | ........ | ........ | ........ | 97.5 | 96.0 | 95.0 |
| * Specific volume at orifice pressure, and quality or superheat, $v_o$ cu. ft. | 5.0 | 14.6 | 27.4 | 52.6 | 99.8 | 228. |
| Area of throat of nozzle $\dfrac{v_o \times W \times 144.}{V_o} = A$ sq. ins. | 3.9 | 11.6 | 27.9 | 50.2 | 97.6 | 227 |

NOTE.—For an expanding nozzle the value of $A$ to be used in computing diameter of turbine is found by multiplying the above value by the expansion ratio of the nozzle.

* In cases of superheated steam in the orifices the values of $v_o$ are to be taken from the curves following page 188.

adiabatically. But all of the energy given up in any one stage does not appear as work delivered by that stage. That which does not appear as work is assumed to be given back to the steam to dry it at constant pressure in case moisture has appeared, or to superheat it, also at constant pressure, in case the steam has not yet become wet. The amount of the reheat (disregarding radiation, conduction, and leakage losses), is represented by $(1-e)h_s$, where $e$ is the stage efficiency and $h_s$ is the heat drop in the stage under consideration.

Let the turbine be required to develop 2000 horse-power at 1000 revolutions per minute, and let the mean bucket-speed be 420 feet per second. The mean diameter of bucket circle will then be

$$D=\frac{420\times 60}{1000\times 3.14}=8 \text{ feet}.$$

Let the initial pressure and degree of superheat be, respectively, 165 pounds absolute and 100° F. Let the pressure in the exhaust pipe be 1 pound absolute or correspond to 28 inches vacuum. From the heat diagram, the heat contents at entrance to the first nozzle-bowls is $H_{B_1}=1252$ B.T.U.; and the final heat contents after adiabatic expansion to 1 pound absolute is $H_2=910$ B.T.U. The available energy, assuming adiabatic expansion, is therefore

$$E=H_{B_1}-H_2=1252-910=342 \text{ B.T.U.}$$

Let the turbine have six stages, and let the energy distribution aimed at be as follows:

First stage, $0.30\,E=h_{s_1}=102$ B.T.U., approximately.
Remaining stages, $0.14\,E=h_{s2}, h_{s_3}$, etc. $=48$ B.T.U., "

Let the stage efficiencies be taken from experimentally determined curves, as 0.48 for the first stage, and 0.55 for each remaining stage.

It is now possible to use a heat-diagram to ascertain the probable steam condition as to pressure, temperature, quality, heat-contents, etc., at the various nozzle-bowls, and to use this information in determining the areas of cross section of the

various sets of nozzles. The method of making calculations is shown below, and the heavy lines in the heat-diagram on the back cover of the book show the calculated expansion curve.

All results in these calculations are in B.T.U. per pound of steam.

| | | |
|---|---|---|
| Initial heat contents, per pound of steam | 1252 | B.T.U. |
| Adiabatic drop in first stage nozzles | 102 | |
| | 1150 | |
| Reheat in first stage $(1-0.48)\times 102$ | 53 | |
| Heat contents at entrance to second stage nozzles | 1203 | |
| Adiabatic drop in second stage nozzles | 48 | |
| | 1155 | |
| Reheat in second stage $(1-0.55)\times 48$ | 22 | |
| Heat contents at entrance to third stage nozzles | 1177 | |
| Adiabatic drop in third stage nozzles | 48 | |
| | 1129 | |
| Reheat in third stage (same as in second stage) | 22 | |
| Heat contents at entrance to fourth stage nozzles | 1151 | |
| Adiabatic drop in fourth stage | 48 | |
| | 1103 | |
| Reheat in fourth stage | 22 | |
| Heat contents at entrance to fifth stage nozzles | 1125 | |
| Adiabatic drop in fifth stage nozzles | 48 | |
| | 1077 | |
| Reheat in fifth stage | 22 | |
| Heat contents at entrance to sixth stage nozzles | 1099 | |
| Adiabatic drop in sixth stage nozzles | 48 | |
| | 1051 | |
| Reheat in sixth stage | 22 | |
| Heat contents of exhaust | 1073 | |
| Heat actually given up, per pound of steam, $1252-1073=$ | 179 | B.T.U. |
| Heat that would have been given up during adiabatic expansion | 342 | " |

Efficiency, $\frac{179}{342}=.524.$

The water-rate based on this efficiency would be $\frac{1,980,000}{179\times 778}=14.2$ pounds.

The efficiencies selected above for the stages have been taken at random, and do not necessarily represent the performance of any particular turbine.

It is to be noted that the final temperature and pressure of the steam, as shown by the expansion curve on the heat-diagram, are slightly above the vacuum conditions assumed. A difference of this kind will always be found in the calculations when the steam in its final condition is moist, and the difference is dependent upon the efficiencies assumed and the heat distribution employed. The efficiencies used in the present example may be assumed to take into account dissipation of energy by shaft friction, windage, leakage, etc., and the calculated water-rate therefore to be in pounds of steam per B.H.P. hour.

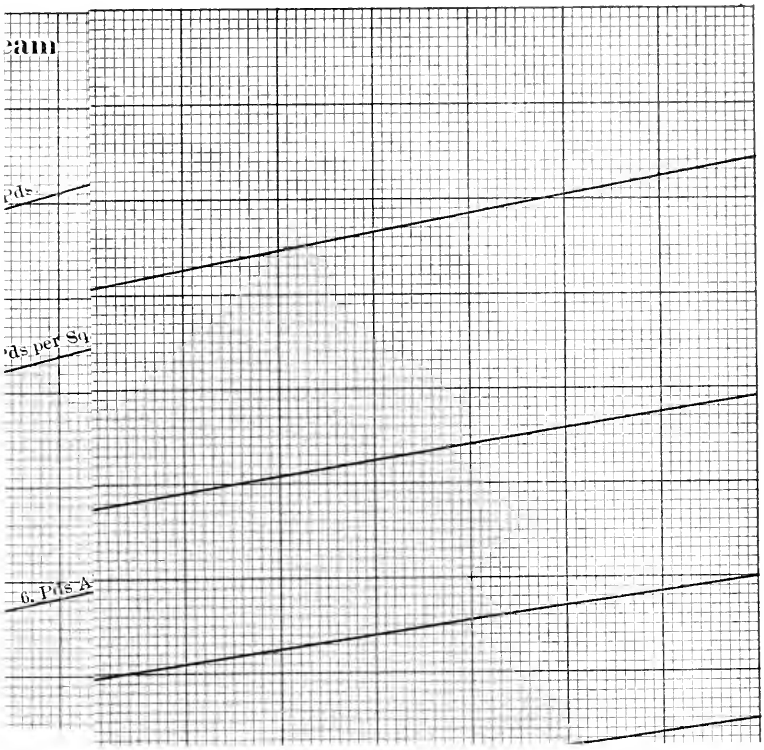
In order to find the height of nozzles and buckets in the last stage, or in fact of any one of the stages, the steps to be taken are similar to those described on pages 170–173. Thus, the area required through the nozzles of the last stage is 227 square inches, to provide for 2000 horse-power. In case it is desired to provide for an overload of 50%, nozzles may be added which can be opened by valves suitably arranged. Supposing it is desired to provide for a possible overload of 50% and that when so overloaded the turbine will require 10% more steam per horse-power hour than is required at normal load, the area of nozzles to be provided must be increased to 375 square inches.

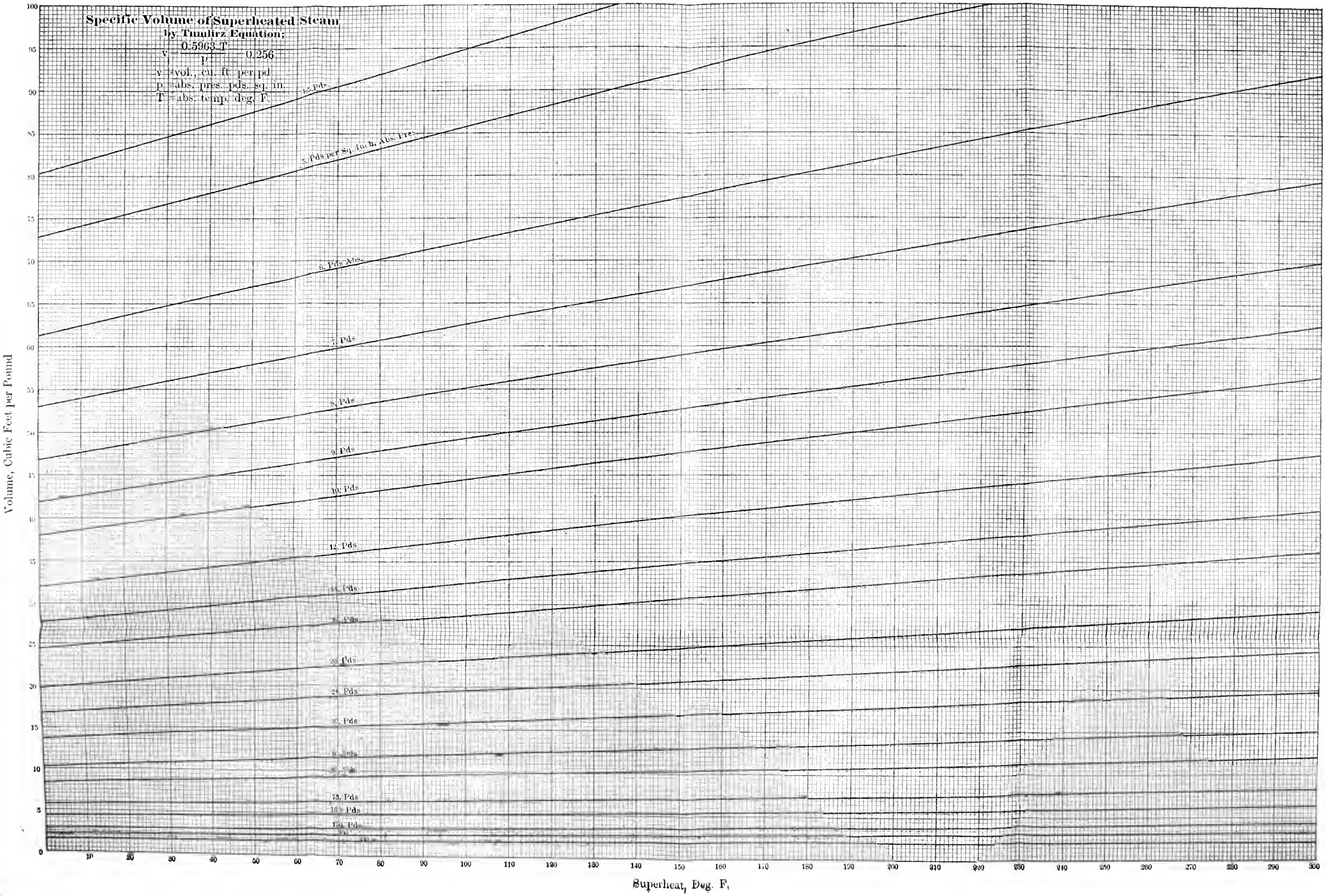
Let the thickness, $t$, of nozzle walls be $\frac{1}{16}$ inch and let the angle of nozzles with the plane of rotation be 25 degrees. ($\sin 25° = 0.423$.) Let the pitch of nozzles be 1.5 inches, and let the nozzles subtend an angle at the center of the shaft of $\Delta = 180°$. Assuming that it is not practicable to make one set of nozzles occupy half the pitch circle, on account of the bolting together of the diaphragm, let the nozzles be made in two sets, each subtending 90° and on opposite sides of the turbine. The height of nozzles in the last stage will then be

$$H = \frac{375}{0.0087 \times 96 \times 0.915 \times 180 \times 0.423} = 6.5 \text{ inches nearly.}$$

It is to be noted, that while the nozzles leading into all the compartments excepting the first are non-expanding, the

velocity of steam from these nozzles is supposed to be that corresponding to the heat-drop from one stage to the next and the nozzle efficiency, and not merely the orifice velocity from which the area is determined. That is, the steam is supposed to be accelerated after it leaves the orifice, or entrance to the straight nozzles. In this connection reference should be made to the experimental work discussed in Chapter VI, where it is shown that for initial pressures less than about 80 pounds absolute the straight nozzle is fully as efficient, if not more so, than the expanding nozzle.


eam
Pds
Pds per Sq
6. Pds A



Specific Volume of Superheated Steam
by Tumlirz Equation;
v = 0.5963 T / p − 0.256
v = vol., cu. ft. per pd.
p = abs. pres. pds. sq. in.
T = abs. temp. deg. F.
5. Pds per Sq. Inch, Abs. Pres.
6. Pds Abs.
7. Pds
8. Pds
9. Pds
10. Pds
12. Pds
20. Pds
25. Pds
30. Pds
75. Pds
Volume, Cubic Feet per Pound
Superheat, Deg. F.

## CHAPTER VIII

### THE IMPULSE-AND-REACTION TURBINE.

As an introduction to the study of the Parsons turbine reference should be made to the descriptive matter on pages 251 to 270, including figures 86 to 100. Before attempting to analyze the turbine on the basis of velocity diagrams, and before taking up the question of frictional resistances, reheat, and the location of the various losses of energy, a simple example will be worked out, assuming adiabatic expansion throughout the turbine.

Let it be decided to design a turbine of 1000 B.H.P., taking steam at 175 pounds absolute pressure per square inch and 160° F. superheat at the throttle, and expanding it to a vacuum represented by 28 inches mercury.

The initial heat contents are found from the heat-diagram to be 1288 B.T.U. per pound, $=H_1$. The final heat contents, after adiabatic expansion to vacuum conditions, are similarly found to be 925 B.T.U. per pound, $=H_2$.

Available energy $=H_1-H_2=363$ B.T.U. per pound.

Let it be decided to make the ratio of peripheral velocity, $u$, to steam velocity, $V$, equal to $\frac{u}{V}=0.60$.*

---

* It should be noted here that if, with a given constant value of $u$ the ratio $\frac{u}{V}$ be increased, the velocity $V$ is necessarily decreased. $V$ varies directly as the square root of the heat given up per stage by the steam, hence there is less and less energy given up per stage as the ratio $\frac{u}{V}$ is increased. Therefore the number of stages necessary in order to absorb a given supply of available energy increases as the ratio $\frac{u}{V}$ increases, for a given constant value of $u$.

Let it be similarly decided to allow a mean peripheral velocity of blades in the first cylinder (see page 217 for definition of "cylinder") of 150 feet per second, and let the values for the second and third cylinders be respectively 240 and 350 feet per second. The steam velocities will then be as shown in the following table:

| Cyl. No. | $u$, Feet per Second. | $V = u \div 0.60$. |
|---|---|---|
| 1 | 150 | 250 |
| 2 | 240 | 400 |
| 3 | 350 | 583* |

* In the third cylinder the velocity will be increased from this value at the first few rows to about 900 feet per second at the last rows, and $\frac{u}{V}$ will decrease to about 0.40.

Let each of the three cylinders absorb energy in the following proportion:

| Cyl. No. | Per Cent. | Amount Absorbed. |
|---|---|---|
| 1 | 25 | $0.25 \times 363 = 91$ B.T.U. |
| 2 | 35 | $0.35 \times 363 = 127$ " |
| 3 | 40 | $0.40 \times 363 = 145$ " |

The heat necessary to be expended in each row of blades in order to give the steam the desired velocity is (since $V_1 = 224\sqrt{H}$.)

$$H = \left(\frac{V_1}{224}\right)^2.$$

| Cyl. No. | $H$ |
|---|---|
| 1 | $\left(\frac{250}{224}\right)^2 = 1.25$ B.T.U. |
| 2 | $\left(\frac{400}{224}\right)^2 = 3.20$ " |
| 3 | $\left(\frac{583}{224}\right)^2 = 6.80$ " |

The number of rows, including both movable and stationary blades, required to absorb the available energy in each stage will then be:

| Cyl. No. | No. of Rows. |
|---|---|
| 1 | $\frac{91}{1.25}=72+$ or 36 in rotor and 36 in casing. |
| 2 | $\frac{127}{3.20}=40$ or 20 in rotor and 20 in casing. |
| 3 | $\frac{145}{6.80}=21+$ or 11 in rotor and 11 in casing. |

If the revolutions per minute are to be 3600, or 60 per second, the mean diameters of cylinders to give the required peripheral velocity will be:

| Cyl. No. | $u$. | Diameter of Mean Blade Circle. |
|---|---|---|
| 1 | 150 | $\frac{150}{3.14\times 60}=0.80$ ft. $=$ 9.6 in.—say 10″. |
| 2 | 240 | $\frac{240}{3.14\times 60}=1.28$ ft. $=$ 15.3 in.—say $15\frac{1}{4}$″. |
| 3 | 350 | $\frac{350}{3.14\times 60}=1.86$ ft. $=$ 22.4 in.—say $22\frac{1}{2}$″. |

The cross-sectional area of the annular space occupied by blades, between the rotor and the casing, is ordinarily made from $2\frac{1}{2}$ to 3 times the net area required for steam flow at the velocity upon which the design is based. This is because the ratio of the area of annular space to the area of exit openings between the blades, is equal to from 2.5 to 3.0. The blade height depends upon this ratio, the volume of steam passing per unit of time, and the velocity of the steam leaving the blades. If the required area at any cross-section is $A$ square

inches, and if the mean diameter of blade-circle at that section is $D$ inches, then

$$\text{Blade height} = \frac{3A}{3.14D} \text{ inches.}$$

Let it be assumed that the steam consumption of the turbine at the rated full load of 1000 B.H.P. is to be 12 pounds per B.H.P.-hour, or that 12,000 pounds of steam are to pass through the blades per hour. This is equivalent to 3.33 pounds per second.

The initial steam pressure at entrance to the blades of the first cylinder is lower than the throttle-valve pressure by perhaps 15 to 25 pounds, because of the wire-drawing effect of the throttle-valve movement, when acted upon by a flyball governor.*

Assuming that in the present case the pressure at entrance to the blades is 150 pounds absolute, and that the steam during its expansion through the throttle-valve follows a constant heat curve (that is, it expands without loss of heat and without doing any work) the temperature will fall from 991 to 987 degrees absolute. Since the temperature of saturated steam at 150 pounds absolute pressure is 819 degrees absolute, the steam at entrance to the blades is superheated by an amount equal to $987-819=168$ degrees.

From the curves of specific volume of superheated steam, opposite page 188, the specific volume at entrance is 3.66 cubic feet per pound.

If adiabatic expansion takes place until the steam shall have given up 91 B.T.U. the pressure at the end of cylinder No. 1 will be approximately 60 pounds absolute, and the temperature 798 degrees absolute. The steam will then be superheated 45 degrees, and its heat contents will be 1197 B.T.U. per pound. The specific volume will be 7.94 cubic feet.

* In some types of Parsons turbine the valve is continually in motion and this wire drawing takes place. In others the valve is either open or closed, and the pressure of the steam is constant under a given load. In the example the former type is assumed.

Since the velocity of steam in the first cylinder is to be 250 feet per second, the necessary cross-sectional area at entrance to and exit from the cylinder, respectively, will be

$$\frac{3.33\times3.66}{250}=0.0488 \text{ sq. ft. or } 7.03 \text{ sq. ins.,}$$

$$\frac{3.33\times7.94}{250}=0.106 \text{ sq. ft. or } 15.2 \text{ sq. ins.}$$

The mean diameter of the blade circle is 10 inches and therefore the blade heights at entrance and exit of the cylinder are, respectively,

$$\frac{3\times7.03}{3.14\times10}=0.672 \text{ inches,}$$

$$\frac{3\times15.2}{3.14\times10}=1.45 \text{ inches.}$$

The heat contents at entrance to the second cylinder is 1197 B.T.U. per pound, and adiabatic expansion is assumed to take place until 127 B.T.U. have been given up. The heat contents will then be $1197-127=1070$ B.T.U., and the pressure after passing the second cylinder will be $10\frac{1}{2}$ pounds absolute (from the heat-diagram). The quality will be 0.92 and the specific volume 34 cubic feet per pound.

Assuming that the steam at entrance to the second cylinder has the same volume as at exit from the first, the cross-sectional area at entrance to the second cylinder (that is, at exit from the first row of blades of that cylinder) should be

$$\frac{3.33\times7.94}{400}=0.066 \text{ sq. ft. or } 9.5 \text{ sq. ins.,}$$

and at exit from the cylinder,

$$\frac{3.33\times34}{400}=0.283 \text{ sq. ft. or } 41 \text{ sq. ins.}$$

The corresponding blade heights at entrance and exit, respectively, are (the mean diameter of cylinder being 15.25 inches)

$$\frac{3\times9.5}{3.14\times15.25}=0.595 \text{ inches,}$$

$$\frac{3\times41}{3.14\times15.25}=2.57 \text{ inches.}$$

In similar manner the blade length at entrance to the third cylinder is found to be

$$\frac{3\times33.7}{3.14\times22.5}=1.43 \text{ inches.}$$

The specific volume of steam at exit from the last row of blades in the turbine is 280 cubic feet per pound, and if the steam velocity should remain constant at 583 ft. per sec. during passage through the blade exits of the last cylinder, the length of blades would become inconveniently great. In the present case it would be about 10 inches. Such length is avoided by increasing the exit-angles of the blades as the last rows are approached, thus allowing the steam velocity to increase rapidly, and permitting the use of shorter blades. Thus, if the velocity should be increased to 900 feet per second, the cross-sectional area required would be 150 square inches, and the blade length

$$\frac{3\times150}{3.14\times22.5}=6.35 \text{ inches.}$$

Summing up, such particulars as have been determined for the turbine would be as follows:

Delivered horse-power at full rated load, 1000.
Revolutions per minute, 3600.
Number of cylinders, 3.
Initial steam pressure 175 pounds absolute at throttle.
Superheat, 160° F. at throttle.
Vacuum, 28 inches.

| Cyl. No. | Mean Diam. Blade Circle. | No. Rows on Rotor. | Peripheral Velocity, Ft. per Second. | Steam Velocity. Ft. per Second. | Heat Drop, B.T.U. | Pressure at Entrance to Cyl., Pds. Abs. | Blade Length at Entrance to Cyl. Inches. | Blade Length at Exit from Cyl., Inches. |
|---|---|---|---|---|---|---|---|---|
| 1 | 10″ | 36 | 150 | 250 | 91 | 150.0 | 0.67 | 1.45 |
| 2 | $15\frac{1}{4}$″ | 20 | 240 | 400 | 127 | 60.0 | 0.60 | 2.57 |
| 3 | $22\frac{1}{2}$″ | 11 | 350 | 583+ | 145 | 10.5 | 1.43 | 6.35 |

It will be shown in the following detailed study of the turbine, how velocity and volume of the steam are affected by frictional and other losses, and how these affect the dimensions of the turbine.

The work done in the first stationary blades of the frictionless reaction-turbine is that necessary to accelerate the

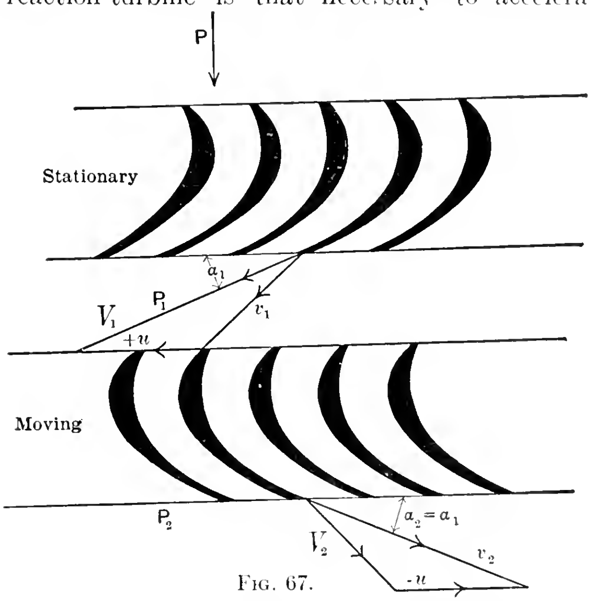



FIG. 67.

jet from its practically zero velocity at entrance to the turbine-casing to the velocity $V_1$ at which it enters the first moving blades. If the work in the stationary blades is called $K_s$, then

$$K_s = V_1^2 \div 2g.*$$

* Per pound of steam.

The velocity relatively to the moving blades, at entrance to them, is $v_1$, and this increases to $v_2$ at exit from the moving blades. The work done in the moving blades is then

$$K_m = \frac{v_2^2 - v_1^2}{2g}.$$

Part of $K_s$ produces pressure against the blades and part is lost as exit energy, due to the velocity $V_2$.

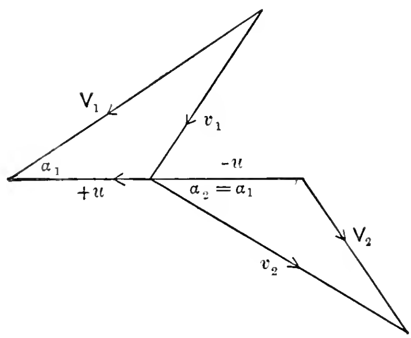



FIG. 68.

The total work, including the energy in the exit steam, is

$$K_t = K_s + K_m.$$

The work done in the moving blades is to the total work done as $\frac{K_m}{K_t}$, and this fraction is called the "degree of reaction."

The net work accomplished upon the turbine is

$$K = K_s + K_m - \frac{V_2^2}{2g},$$

expressed in foot-pounds, and the efficiency is

$$K \div (K_s + K_m).$$

*Example No. 1.*—Let initial steam-pressure = 150 pounds per square inch absolute; let the drop in pressure in the first set of guide-blades be 10 pds. and let a similar drop occur in the first set of moving blades. Let $\alpha_1 = 30°$ and let $\alpha_2 = \alpha_1$. Find the work done on the moving blades per pd. steam. Let peripheral velocity = 250 ft. per second.

It is first necessary to calculate the velocity at entrance to the moving blades. Assume the expansion to be adiabatic, with no frictional losses.

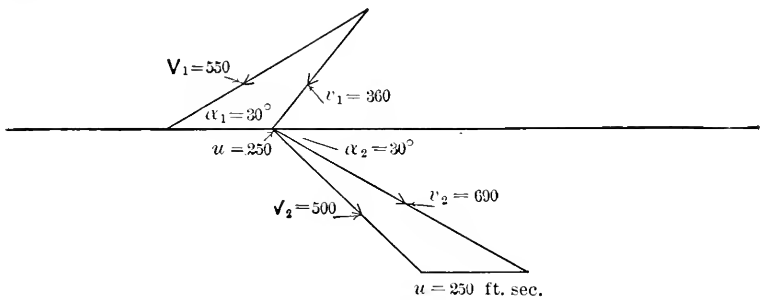



FIG. 69.

Heat given up in guide-blades = 6.0 B.T.U. or $V_1 = 550$ ft. per second. From the velocity diagram, $v_1 = 360$ ft. per second. The work done in the moving blades is $\frac{v_2^2 - v_1^2}{2g}$ foot-pds., which equals the heat given up during the passage of the steam through the moving blades multiplied by 778. The heat given up during adiabatic expansion from 140 to 130 pds. is 6.9 B.T.U.

Then
$$\frac{v_2^2 - v_1^2}{2g} = 6.9 \times 778,$$

or

$$v_2 = \sqrt{64.4 \times 6.9 \times 778 + (360)^2} = \sqrt{475{,}312} = 690 \text{ ft. per sec.},$$

approximately. The work done in the stationary blades is

$$K_s = \frac{V_1^2}{2g} = \frac{(550)^2}{64.4} = 4700 \text{ ft.-pds. per sec.}$$

The work done in the moving blades is

$$K_m = \frac{v_2^2 - v_1^2}{2g} = \frac{(690)^2 - (360)^2}{64.4} = 5360 \text{ ft.-pds. per sec.}$$

The total work done is

$$K_t = K_s + K_m = 4700 + 5360 = 10,060 \text{ ft.-pds. per sec.}$$

From this last is to be deducted the work

$$\frac{V_2^2}{2g} = \frac{(500)^2}{64.4} = 3890 \text{ ft.-pds. per sec.}$$

The net work accomplished in the turbine is

$$K = K_s + K_m - \frac{V_2^2}{2g} = 6170 \text{ ft.-pds. per sec.,}$$

or 11.2 horse-power.

The efficiency is

$$\frac{K}{K_s + K_m} = \frac{6170}{10,060} = 61.7\%.$$

The degree of reaction is

$$\frac{K_m}{K_t} = \frac{5360}{10,060} = 0.536,$$

or approximately one-half degree reaction, which is about that used in reaction-turbine construction. The exhaust energy in the above turbine is so high that the steam consumption would be very large, and the need for more stages is obvious. Thus the steam consumption per horse-power per hour would be $\frac{3600}{11.2} = 321$ pounds for a turbine of only one stage.

The many-stage reaction-turbine consists of guide and rotating rows of blades, as indicated in Fig. 70. There may be many consecutive rows, all having the same diameter, followed by others of greater diameter, as the required area for passage of the steam becomes greater and greater.

Assuming that the diameter of the rows, or wheels, is con-

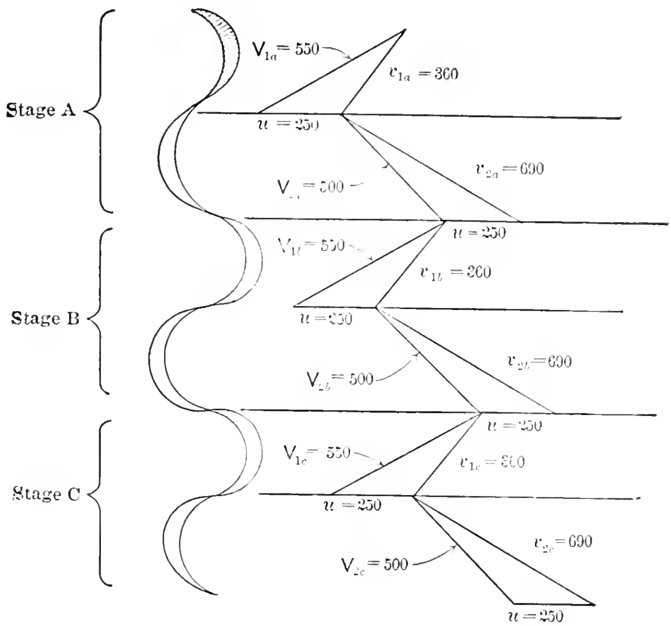



Fig. 70

stant, the peripheral velocity of all blades will be the same Let this be called $u$, as before.

The diagram, Fig. 70, is constructed for constant conditions of absolute and relative velocity throughout the various stages, and, as stated above, all stages are alike in diameter. In the first guide-wheel the work done is

$$V_1^2 \div 2g.$$

In the first moving wheel the work done by expansion of the steam is that due to increase of velocity from $v_1$ to $v_2$ and equals

$$\frac{v_2^2 - v_1^2}{2g}.$$

In each guide-wheel after the first the work done is due to increase from $V_2$ to $V_1$, and the kinetic energy thus produced and then applied to the succeeding moving blades equals

$$\frac{V_1^2 - V_2^2}{2g};$$

and the work in each moving wheel is the same as stated above; that is,

$$\frac{v_2^2 - v_1^2}{2g}.$$

In general, if there are $n$ sets of wheels (that is, $n$ stages), each consisting of one guide and one moving wheel, there will be $n-1$ sets, or stages, besides the first stage. The total work including that of the first stage will be

$$\frac{V_1^2}{2g} + \frac{v_2^2 - v_1^2}{2g} + (n-1)\left[\left(\frac{V_1^2 - V_2^2}{2g}\right) + \left(\frac{v_2^2 - v_1^2}{2g}\right)\right].$$

Let $K$ = the work in each stage except the first, so that

$$K = \frac{V_1^2 - V_2^2}{2g} + \frac{v_2^2 - v_1^2}{2g}.$$

The efficiency of a single stage may be found as follows:

If $\alpha_2 = \alpha_1$, as in Figs. 67 and 68, then $V_1 = v_2$, and $V_2 = v_1$.

$$K = 2\left(\frac{V_1^2 - v_1^2}{2g}\right) \text{ and efficiency} = K \div \frac{2V_1^2}{2g} = \frac{(V_1^2 - v_1^2)}{V_1^2}.$$

But from Fig. 62, by trigonometry,

$$v_1^2 = V_1^2 + u^2 - 2uV_1 \cos \alpha.$$

Therefore the efficiency $= \frac{2u \cos \alpha}{V_1} - \frac{u^2}{V_1^2}$

$$= \frac{u}{V_1}\left[2 \cos \alpha - \frac{u}{V_1}\right].$$

The variation of efficiency for $V_1 = 300$ feet per second, with variation of $\alpha$ and $u$, is shown on page 228.

The total work done in $n$ stages is

$$K + (n-1)K + \frac{V_2^2}{2g} = nK + \frac{V_2^2}{2g}.$$

Since $\frac{V_2^2}{2g}$ is the work lost at exit from the turbine, the net work done, per pound of steam, $= nK$.

*Example No. 2*—Taking the velocities as given in Fig. 70.

$V_1 = 550$ for each guide-wheel,
$V_2 = 500$ for each guide-wheel,
$v_1 = 360$ for each moving wheel,
$v_2 = 690$ for each moving wheel,

$$K = \frac{(550)^2 - (500)^2}{64.4} + \frac{(690)^2 - (360)^2}{64.4} = 6170 \text{ ft.-pds.},$$

work done in each stage per pound of steam used. This is, of course, for a frictionless and otherwise ideal turbine. In such a machine, if expansion occurred from 150 pds. abs. to 130 pds. as in the example on page 197, there would be available 12.9 B.T.U. or, approximately, 10,000 foot-pounds of energy per pound of steam, and an ideal turbine would require only $\frac{10,000}{6170} = 1.62$ stages to completely utilize the energy available.

If expansion should occur from 150 pds. to 1.5 pds. absolute, as in the example on page 83, there would be 290 B.T.U. available, or $290 \times 778 = 225,000$ foot-pounds. In an ideal

reaction-turbine of the kind above descr.bed, the number of stages required to absorb this energy would be

$$\frac{225,000}{6170}=36, \text{ approximately.}$$

**Action of the Steam upon the Buckets.**—The guide-blades act as nozzles leading to the moving-blades. Under normal conditions the guide-blades receive steam from the preceding moving-blades at an absolute velocity, or velocity with respect to the earth, of $V_2$ (Fig. 68), and discharge it upon the succeeding moving-blades at velocity $V_1$, larger than $V_2$. Considered with respect to the motion of the moving-blades, the velocity of the entering steam is $v_1$, and during its passage through the moving-blades the steam has its velocity relatively to the moving-blades increased to $v_2$. The total work done upon the row of moving-blades is that due to the following two causes: First, impulse, as the energy $(V_1^2-V_2^2)\div 2g$ per pound of steam, produced in the guide-blades, is expended upon the moving-blades; and, second, the reaction accompanying the change in the moving-blades from $v_1$ to $v_2$, and resulting in an energy expenditure upon the blades of $(v_2^2-v_1^2)\div 2g$ per pound of steam used. The guide- and moving-blades are ordinarily approximately alike as to angles, and when this is so, half the work is due to impulse and half to reaction, provided that the heat-drop is the same in the two rows. If $V_2$ should equal $V_1$, as in Fig. **71**, the work would be due entirely to reaction, and equal to $(v_2^2-v_1^2)\div 2g$. If $v_2'$ should equal $v_1$, the total work would be due to impulse, and equal to $(V_1^2-V_2'^2)\div 2g$, just as in the impulse-turbine. If $v_2''$ should equal $v_1$, and $V_2''=V_1$, as shown in dotted lines, the blade would no longer be curved, but would have the outline $AB$ (Fig. 71) and the work done would be zero.

**Losses of Energy in the Turbine** may be classified as follows:

(*a*) The effect of friction between the steam and the metallic walls and moving parts of the turbine, which is to cause the exhaust from a given stage to carry away more heat-energy

than it would in a frictionless conducting-channel. The cause of this is described in Chapter V. A similar result is brought

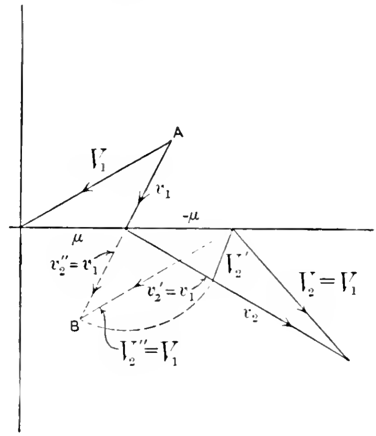



Fig. 71.

about by the friction due to eddies in the steam. The "curve of frictional effect" (page 219). is useful in representing the variation of this source of loss, but it is by no means certain that the friction loss varies according to the curve in Fig. 77. Examples 4 and 5 assume the loss to be constant, corresponding to a value of $y=0.26$.

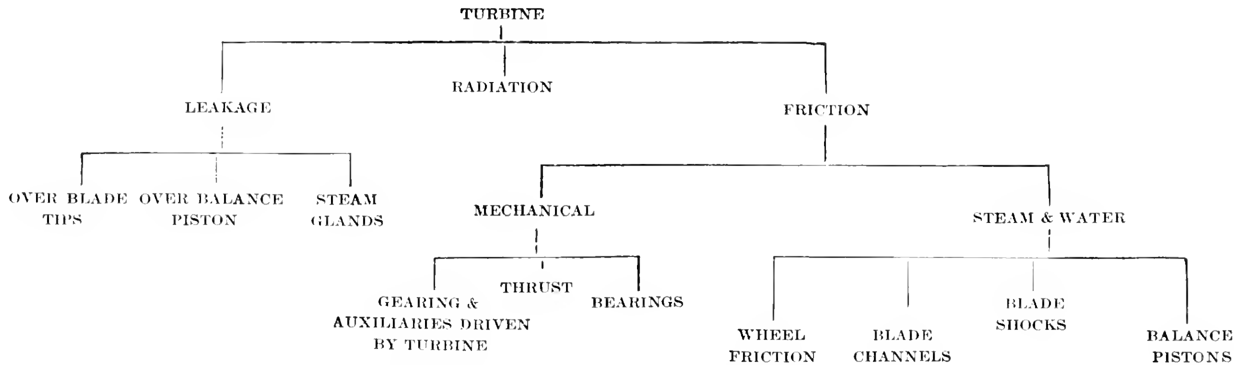
(*b*) Resistance to movement of the rotating parts in the atmosphere of steam within the turbine casing, called "windage." This causes a frictional loss, and its effect is probably greatest at the high-pressure end of the turbine, diminishing as the low-pressure end is approached.

(*c*) Mechanical friction in journal-bearings, glands, and stuffing-boxes.

(*d*) Leakage losses through clearance-spaces, glands, etc.

(*e*) Radiation losses.

**The steam consumption of a turbine** working between known limits may be calculated as follows, for assumed losses due to steam friction and friction of rotating parts, and loss due to leakage. The dotted line, Fig. 72, indicates the condition of the steam during expansion through the turbine.




Distribution of Losses in the Parsons Turbine.

First, assuming adiabatic expansion, allowing for no losses:

| | | | |
|---|---|---|---|
| Heat of liquid at 165 pds. abs. . . = 337.6 | | | 337.6 |
| Heat of vapor'n at 165 " " . . = 855.3 | $855.3 \times 0.98$ | | = 838.2 |
| | | | 1175.8 |
| Heat of liquid at 1.23 pds. abs. . . = | | 77.06 | |
| Heat of vapor'n at 1.23 " " . . = 1037.8 | | | |
| | $1037.8 \times 0.763$ | = 791.84 | |
| | | 868.90 | 868.90 |
| Heat given up $= H_1 - H_2 =$ . . . . . . . . . . . . . | | | 306.9 B.T.U. |

Let the loss of heat in the blades, due to friction, correspond to $y = .26$, or $(1-y)(H_1 - H_2) = 306.9 \times 0.74 = 227$ B.T.U. Suppose the energy in the exhaust is 4% of the initial energy

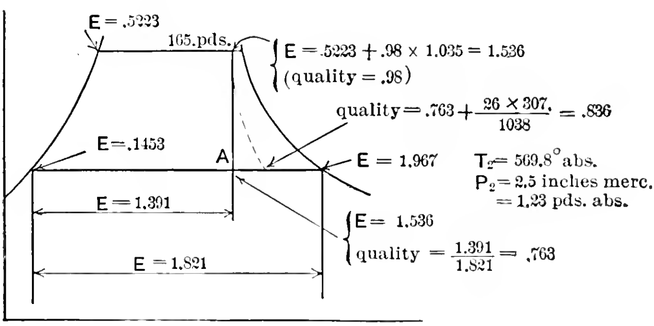



FIG. 72.

minus loss in the blades, and the loss due to friction of the rotating drum and of the bearings $= 14\%$. Let the loss due to leakage $= 7\%$. Sum of losses $= 25\%$, besides the 26% heat loss due to steam friction. Then $75\% \times 227 = 170$ B.T.U. available from each pound of steam flowing through the turbine. Suppose 1 pd. flows per sec., or 3600 pds. per hour. Ft.-pds. per min. $= 60 \times 170 \times 778 = 7{,}935{,}600$. Horse-power $= \dfrac{7{,}935{,}600}{33{,}000} = 240$. Steam consumption $= \dfrac{3600}{240} = 15$ pds. per delivered horse-power hour.

**In reaction-turbine design the assumptions made at the start** are chosen from among the following items:

1. Initial and final steam-pressures.
2. Initial quality of steam, or degree of superheat, if any.
3. Losses to be experienced by the steam during its passage through the machine.
4. Initial velocity of steam as it leaves the guide-blades.
5. Angle at which the guide-blades discharge to the moving blades.
6. Angle at which the moving blades discharge to the guide-blades.
7. Peripheral velocity of the blades in the various cylinders.

Assumptions as to the above make it possible to determine cross-sectional areas at different points in the turbine, the length and width of blades, and to estimate the heat losses. From these data the probable steam consumption may be calculated for a given rate of power developed.

The number of revolutions per minute may be decided upon from considerations of the use for which the turbine is intended. The drop in energy in the various stages determines the initial velocity of steam through the blades, and the peripheral velocity of the latter is usually from one third to two thirds or more of the steam velocity, the ratio for highest efficiency depending upon the exit angles of the blades.*

From Fig. 73 it is obvious that the cross-sectional area through a row of blades decreases as the exit angle with the direction of motion of the blade becomes smaller. Considering the two extreme limiting cases, if the steam were discharged from a set of blades in the direction of motion of the blades—that is, if $\alpha$ became zero—the cross-sectional area would become zero. If the blades discharged in an axial direction, the cross-sectional area, assuming infinitely thin blades, would be equal to the length of the blades, multiplied by the circumference on the mean diameter of the row of blades. That is, the area would equal the whole annular area swept by the blades.

* In electrical work a ratio of about 0.6 has been frequently used. See page 226 for further data.

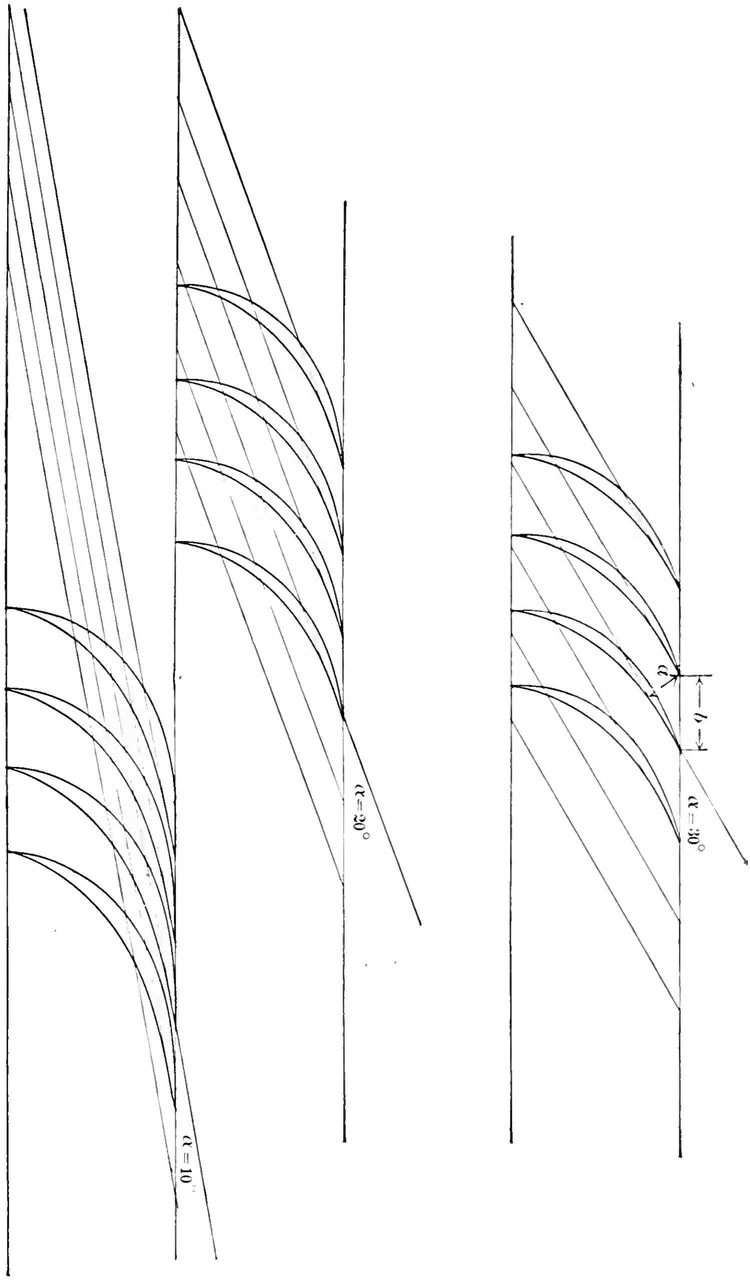



FIG. 73.

Between these two extremes are the actual conditions, and the actual area for infinitely thin blades would be to the whole annular area swept by the blades as $a$ is to $b$ (Fig. 73). But $a \div b = \sin \alpha$, and the area for blades having a length $L$ and rotating on a mean diameter $D$ would be

$$\text{area} = \pi DL \sin \alpha.$$

The blades have a certain thickness to afford strength, and the diameter of cylinder and length of blades must be proportioned so that the required area for passage of steam may be obtained. If the thickness of the blades is half the mean clear opening between two blades, then the area corresponding to blades without thickness should be multiplied by 1.5, and the proper diameter of cylinder and length of blades calculated. The blade thickness must in all cases be taken account of in calculating cross-sectional area.

Fig. 73 shows how the area decreases with the angle $\alpha$, and that for a given axial space occupied by a row of blades the steam-channel becomes longer, as well as narrower, with decrease of the angle $\alpha$. From these facts it follows that, while the power absorbed per stage apparently increases as $\alpha$ decreases, thus reducing the number of stages, the friction losses become greater as $\alpha$ decreases. There is thus a limit beyond which it does not pay to decrease the exit angles. In reaction-turbines the exit angle is ordinarily from 20° to 30° for both guide and moving blades. If the exit angle is made too large, each stage absorbs and delivers too little energy. and too many stages are required. This not only increases the size of the turbine, but also lengthens the path of the steam and makes the friction losses greater.

It has been shown that the friction losses increase with the square of the velocity of the steam. Making the drop in each wheel small, by increasing the exit angles, results in increased number of stages, and large friction losses, due to the lengthened steam path. Making the drop large in each stage increases the velocity of steam, and the friction losses increase as the square of the velocity. The choice of the con-

ditions as outlined at the top of page 206 is to be made with a view to reducing friction and other losses to a minimum, by properly proportioning, with respect to each other, the peripheral velocity of the blades of the various stages, the angles of exit, and the drop in heat contents per stage, which determines the velocity of the steam.

**In making calculations based on the preliminary assumptions,** the heat diagram is used. Considering only that portion to the left of the curve of saturated steam, curves of constant heat, constant quality, and constant volume are drawn, and methods of interpolation may be used for finding the quality, volume, and heat contents of steam at any temperature and specific entropy within the limits of the diagram.

The intervals between all quality curves are alike for any one temperature, and the same is true of the curves of constant heat.

*Example No. 4.*—Let steam at 165 pds. abs. and 98% quality expand to a pressure of 2.5 inches of mercury, or 1.23 pds. abs. The upper and lower temperatures are 826.5 and 570 degrees absolute respectively.

(*a*) Assuming that expansion is *not* adiabatic, but that the steam loses 26% because of friction, find what the quality of the steam will be at the lower temperature, and at 600, 650, 700, 750, and 800 degrees absolute. See dotted expansion line, Fig. 72. Note also Fig. 26 and discussion in Chap. V.

(*b*) Find the heat contents, per pound of the steam, at each of the above temperatures.

(*c*) Plot curves of heat drop, and of specific volume of the steam, for the expansion indicated, as is done on page 211. To find the volumes, multiply the specific volume of dry steam at the various temperatures by the corresponding qualities.

By plotting an expansion curve on the heat diagram using the qualities found in (*a*) the curve of "heat-given-up" may at once be derived. Use of the tabular form given on p. 202 will greatly simplify and facilitate calculation.

The curves asked for in the above example, and plotted in Fig. 74, represent the characteristics of the turbine, giving for each cylinder the mean values of peripheral and steam velocity, and mean cross-sectional areas for passage of steam. These are tabulated below. In designing it is advisable to plot the curves as the first step, and calculate the remaining quantities in the table from the curves as a basis.

Let the peripheral velocity of the turbine-blades vary as shown on the curve (Fig. 74), and let the relation between peripheral and steam velocities at entrance to the moving blades be as follows:

$$\frac{u}{V_1}=0.35.$$

From this and the curve of peripheral velocity the curve of $V_1$ may be plotted. The curve of peripheral velocity is assumed so that a satisfactory length may be given the turbine-blades. The blades of the first cylinder should not be excessively short, otherwise the clearance would be too great a percentage of the total cross-sectional area. A high peripheral velocity means high initial steam velocity, which in turn means small cross-sectional area for passage of steam, and consequently short blades. A certain amount of clearance space is necessary for mechanical reasons, and steam is free to leak through without doing work. If the blades are very short, the clearance becomes a considerable percentage of the total cross-sectional area for passage of steam, and leakage is excessive. For this reason the initial peripheral and steam velocities are kept low and the blades made correspondingly long.

From these considerations the curve of peripheral velocity begins at about 130 ft. per second in the present case and gradually increases to about 350 ft. per second. The corresponding initial steam velocity curve begins at 360 and ends at about 1040 ft. per second, between the limits of temperature 826 and 570 degrees absolute. The relation between these two curves is $V_1=u\div0.35$.

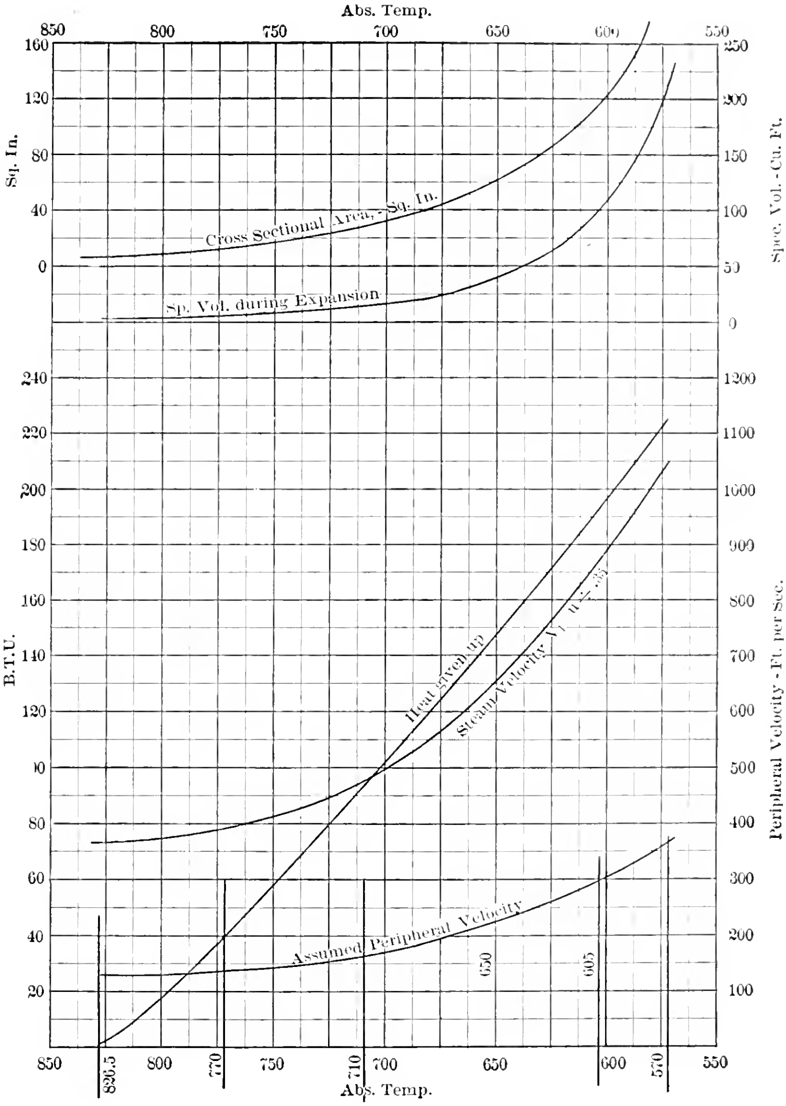

Abs. Temp.
850
800
750
700
650
600
550
160
120
80
40
0
Sq. In.
250
200
150
100
50
0
Spec. Vol. - Cu. Ft.
Cross Sectional Area, - Sq. In.
Sp. Vol. during Expansion
240
220
200
180
160
140
120
100
80
60
40
20
B.T.U.
1200
1100
1000
900
800
700
600
500
400
300
200
100
Peripheral Velocity - Ft. per Sec.
Heat given up
Steam Velocity
Assumed Peripheral Velocity
650
605
826.5
770
710
570
Abs. Temp.

FIG 74.

Let the turbine consist of five cylinders, the blades being of a single length for each cylinder. The cylinders will be arranged to absorb the heat drop between the temperature lines shown in Fig. 74, and in the following table, which contains various necessary quantities taken from the curves, or calculated as described:

| Cylinder No. | Degrees Temp. Drop. | B.T.U. Heat Drop. | Mean Peripheral Vel. $u$. | Mean Initial Vel. $V_1$. | Mean Sp. Vol. of Steam. | Mean Diameter of Row of Blades, Ins. | Average Cross-sectional Area of Cylinder, Sq. In. | Length of Blades, Ins. |
|---|---|---|---|---|---|---|---|---|
| 1 | 56 | 43 | 130 | 375 | 4.03 | 16.5 | 6.5 | .50 |
| 2 | 60 | 53 | 150 | 420 | 9.02 | 19.0 | 13.0 | .875 |
| 3 | 60 | 53 | 200 | 550 | 24.80 | 25.5 | 27.3 | 1.37 |
| 4 | 45 | 46 | 270 | 750 | 66.80 | 34.5 | 53.8 | 2.00 |
| 5 | 35 | 32 | 330 | 950 | 164.00 | 42.0 | 103.0 | 3.14 |

Let the turbine be required to develop 1000 brake horsepower, at 1800 revolutions per minute or 30 per second. This fixes the mean diameter of the rows of blades of the various cylinders, since the peripheral velocity is determined by the curve. Let this be calculated and inserted in the table as above. The mean specific volume of the mixture of steam and water may be found from the curve at the top of Fig. 74, for each cylinder. The curve is to be found, in the first place, from the steam-table values of specific volume of dry steam, by multiplying these by the quality of steam as determined from the expansion curve on the diagram. Each cylinder of a turbine is usually made to gradually enlarge in cross-sectional area as the steam expands. The present calculations apply to the average cross-section for each cylinder.

To find the cross-sectional areas it is necessary to calculate or to ascertain in some way the probable steam consumption of the turbine, as the volume of steam flowing per second is required. From the expansion curve on the heat diagram, the heat given up per pound of steam may be found by measurement. In this case it is the same as the quantity found on page 205, or 227 B.T.U. Let the other losses be the

same as found on page 205, which result in a steam consumption of 15 pounds per brake horse-power hour.

To calculate the cross-sectional area for each cylinder let v = mean specific volume of the steam and water mixture at the cylinder under consideration. The weight flowing per second when developing 1000 horse-power will be $\frac{15,000}{3600}=4.2$ pounds, and the volume per second will be $4.2\times v$. Taking the velocity as that at exit from the guide-blades, the cross-sectional areas for the first and the succeeding cylinders will be:

$$A_1=\frac{4.2\times 4.03}{375}=0.045 \text{ sq. ft.}= \quad 6.5 \text{ sq. ins.}$$

$$A_2=\frac{4.2\times 9.02}{420}=0.090 \text{ `` `` }= \quad 13.0 \text{ `` ``}$$

$$A_3=\frac{4.2\times 24.8}{550}=0.190 \text{ `` `` }= \quad 27.3 \text{ `` ``}$$

$$A_4=\frac{4.2\times 66.8}{750}=0.374 \text{ `` `` }= \quad 53.8 \text{ `` ``}$$

$$A_5=\frac{4.2\times 164}{950}=0.715 \text{ `` `` }= 103.0 \text{ `` ``}$$

These are inserted in the table above.

To find the length of blades in the present problem assume that the exit angle for all the stages is 22 degrees.

As shown on page 208, if $L$ is the length of blade, and $D$ the mean diameter of section of the annular space occupied by blades, the net area for passage of steam would be, for infinitely thin blades,

$$A=\pi DL \sin\alpha, \quad \text{or} \quad L=\frac{A}{\pi D \sin\alpha}.$$

If the blades, due to their thickness, occupy one third of the cross-sectional space, the necessary area becomes $1.5A$, or, since $\sin 22^\circ = 0.374$,

$$L=\frac{1.5A}{3.14D\times 0.374}=\frac{1.28A}{D}.$$

Calculating the mean length of blade, the following values are obtained:

$$L_1 = \frac{1.28 \times 6.5}{16.5} = 0.50 \text{ inch.}$$

$$L_2 = \frac{1.28 \times 13}{19} = 0.875 \text{ ''}$$

$$L_3 = \frac{1.28 \times 27.3}{25.5} = 1.37 \text{ ''}$$

$$L_4 = \frac{1.28 \times 53.8}{34.5} = 2.00 \text{ ''}$$

$$L_5 = \frac{1.28 \times 103}{42} = 3.14 \text{ ''}$$

These have been inserted in the table on page 212.

It now remains to determine the number of stages that are necessary in each cylinder to absorb the energy given up during the fall in temperature assumed at the beginning of the problem.

From Fig. 75, Plate XIII, it is evident that if $\alpha_1 = \alpha_2$, $V_{1a}$ and $v_{2a}$ will be equal to each other. Also, $v_{1a}$ will equal $V_{2a}$.

The energy given up per stage in any cylinder is equal to the sum of the amounts given up in a row of guide-blades and a row of moving blades and equals, for each stage of the first cylinder in the present example,

$$K = \frac{V_{1a}^2 - V_{2a}^2 + v_{2a}^2 - v_{1a}^2}{2g}.$$

But $\qquad V_{1a}^2 = v_{2a}^2,$

and $\qquad V_{2a}^2 = v_{1a}^2.$

$$\text{Therefore } K = \frac{2(V_{1a}^2 - v_{1a}^2)}{2g} = \frac{2(V_{1a}^2 - V_{2a}^2)}{2g}.$$

This simply means that, under the stated conditions (equal exit angles), the energy given up in a rotating wheel equals

ıge

‘

‘

‘‘

ec.

2 d

$V_{1a}$ = Absolute Velocity Leaving Guide Blades, Stage $a$
$v_{1a}$ = Relative " " " " " $a$
$V_{2a}$ = Absolute " " Moving " " $a$
$v_{2a}$ = Relative " " " " " $a$
Similar Notation for Stages $b$, $c$, $d$, and $e$

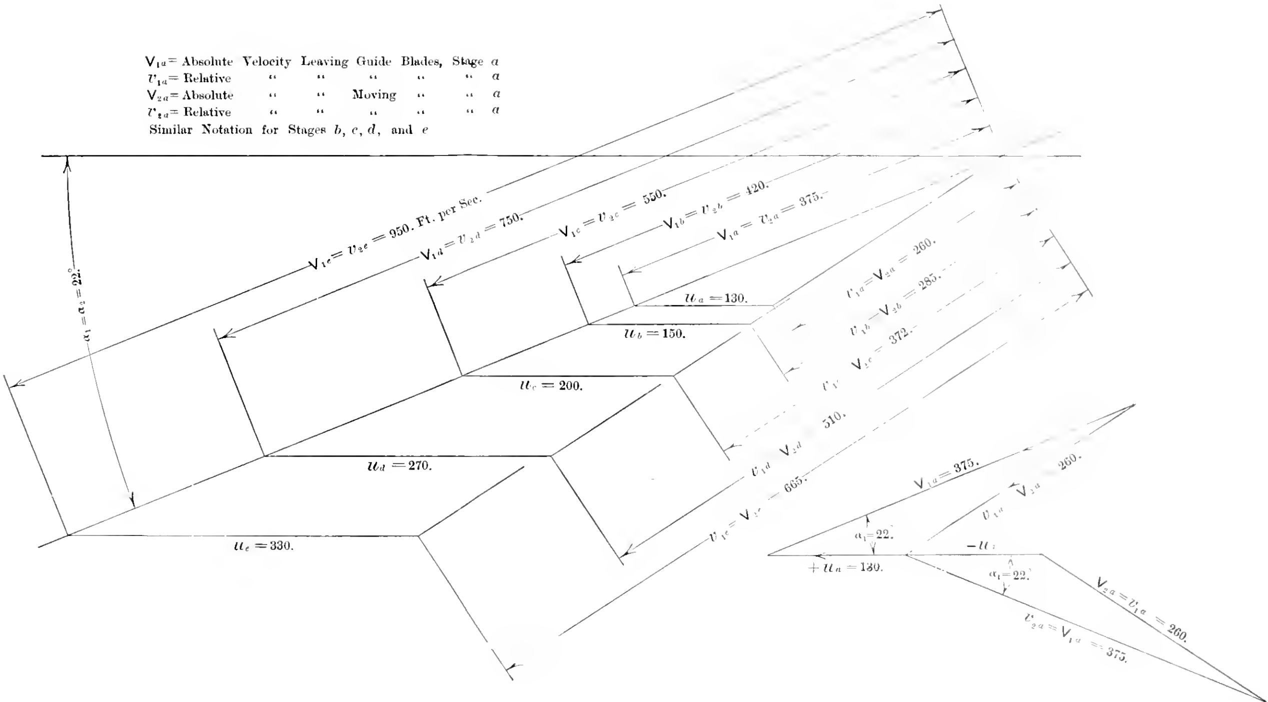



FIG. 76



FIG. 75.

[To face p. 214.

that given up in the guide-wheel before it. It is necessary to construct the velocity diagram for only one of the wheels or rows of blades in a cylinder, since that for the others would be exactly similar.

In Plate XIII the single line making the angle $\alpha = 22°$ with the direction of motion of the blades may be used to represent in direction all the initial velocities, and combining them with their respective peripheral velocities gives the relative velocities necessary for finding the value of $K$ in the above equation. The values of $V_1$ and $V_2$ tabulated on page 212 were taken from the velocity diagram on Plate XIII. The work done in each stage of each cylinder, and the number of stages required to absorb the energy given up in each cylinder, may be calculated as follows: Taking the first cylinder, each stage absorbs the work

$$K_a = \frac{2(V_{1a}^2 - V_{2a}^2)}{2g} = \frac{2[(375)^2 - (260)^2]}{64.4} = 2270 \text{ ft.-pds.}$$

or 2.91 B.T.U.

Since there are 42 B.T.U. to be absorbed during the passage of the steam through this cylinder, the number of stages required will be

$$\frac{42}{2.9} = 14.5, \text{—say 15 stages.}$$

Similar calculations give the following number of stages for each cylinder.

| No. of Stage. | Stages. |
|---|---|
| 1 | 15 |
| 2 | 14 |
| 3 | 8 |
| 4 | 4 |
| 5 | 2 |

These figures, as they stand, would not be satisfactory for use in determining the final dimensions of a turbine. The angles of the blades would be varied more or less from one cylinder of blades to another, to suit various requirements, and the cylinders would usually not be so numerous as here indicated. For a turbine of the size given, three cylinders might be used, leaving the first two about as the figures indicate, but rearranging the last three cylinders so as to combine them into one, consisting of blades increasing in size as they approach the exhaust end. By a series of calculations similar to those above, the required variations may be determined.

The pressure drops in the above example are, approximately:

| | | | |
|---|---|---|---|
| 1st cylinder | ...... | 165 pds. abs. to | 76 pds. abs. |
| 2d " | ...... | 76 " " " | 29 " " |
| 3d " | ...... | 29 " " " | 9 " " |
| 4th " | ...... | 9 " " " | 3.2 " " |
| 5th " | ...... | 3.2 " " " | 1.2 " " |

This large number of cylinders was adopted in order to give practice in making the calculations, but a better arrangement from both thermal and mechanical considerations, would result from the following conditions:

*Example No. 5.*—Proportion an impulse-and-reaction turbine according to the curves given in Fig. 74 with the following pressure drops:

| Cylinder No. | Pressure Drop. |
|---|---|
| 1 | ........165 pds. abs. to 50 pds. abs. |
| 2 | ........ 50 " " " 16 " " |
| 3 | ........ 16 " " " 1.2 " " |

The following table gives the particulars taken from the curves on page 211. In everything except pressure drop the particulars of the design are the same as for the turbine of five cylinders, worked out above. From the curves, page 211, and the temperature drop corresponding to the fall in pres-

sure, the quantities are calculated as was done in the previous example.

| Cylinder No. | Pressure Drop, Pds. | Temperature Drop, Degs. | Heat Drop, B.T.U. | Mean Velocities. | | | Mean Specific Volume Cu. Ft. | Mean Diameter Blades, Ins. | Mean Clear Area, Sq. In. | Mean Len'th Blades Ins. | Number of Stages |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Peripheral. | Steam $V_1$. | Steam $V_2$. | | | | | |
| 1 | 115. | 84 | 68 | 138 | 385 | 260 | 6 | 17.5 | 9.4 | 0.69 | 21 |
| 2 | 34 | 65 | 59 | 170 | 435 | 325 | 18 | 20.5 | 22.7 | 1.42 | 18 |
| 3 | 14.8 | 107 | 100 | 265 | 775 | 535 | 100 | 34.0 | 78.0 | 2.94 | 8 |

Each of the sections of the turbine (three in this case) is properly called a *cylinder*, and each cylinder contains blades of various lengths, increasing as the pressure becomes lower, thus affording increasing area for the passage of steam. Each cylinder may contain several rows of blades of given length and contour, and then several rows of another length and contour. Each of these sets of rows is called a *barrel*, and a cylinder may contain many barrels. The figures in the table refer to the *mean* dimensions of the respective cylinders, hence the blades at the high-pressure end of the cylinder will be shorter and those at the low-pressure end longer than given in the table.

**The diameter of the spindle,** or rotating part of the turbine, depends primarily upon the peripheral velocity of blades and the rate of revolution; the former depends upon the initial steam velocity employed. With given diameters for the various cylinders, and with certain thickness, spacing, and exit angles of blades, the length of blades depends upon the necessary cross-sectional area for the passage of steam through the various cylinders.

The Parsons type of turbine for stationary use has ordinarily three cylinders, and the mean diameters of the rows of blades in the various cylinders are made in about the following proportion:

$d$ = mean diameter of smallest cylinder;
1.4 to 1.5$d$ = " " " middle "
2 to 2.75$d$ = " " " large "

Calling one stationary and one moving row of blades, taken together, a stage, there are ordinarily from 50 to 100 or more stages in turbines of fairly large size; that is, from 300 K. W. upward.

Since turbines are used in connection with very low exhaust-pressures, the volume of steam passing the low-pressure blades per second becomes very great, and the diameter and blade dimensions for that end of the machine should be considered first. The dimensions of the smaller parts may then be proportioned accordingly.

**Variation of Friction Loss.**—The experimental work discussed in Chapter VI indicates that the friction losses in an expanding nozzle increase as the pressure decreases, and that the increase of the value of $y$ is very rapid at very low pressures. Experiments with turbines show that much more energy is lost if the steam used is moist than is lost when dry or when superheated steam is used. During expansion the steam gives up heat, as work, and a considerable amount of water of condensation is formed. The presence of water in the steam is thought to be responsible for what cutting of the blades occurs, and this indicates that the water causes resistance to passage of the steam.

From these indications it has sometimes been considered that the friction loss, as represented by $y$, is greater in extent towards the low-pressure end of the machine than it is at early points of the steam-path, and the "Curve of frictional effect" shown in Fig. 77 has been drawn with this point in mind. It shows the value of $y$ to be used at each of the temperatures dealt with in designing the turbine. The flattening out of the "Curve of heat given up" shows that if the losses due to the accumulation of water on the blades, or along the steam-path, should increase according to the curve assumed, it would not pay to reduce the temperature of the exhaust below 540°. As the use of superheated steam reduces the losses in the turbine, it seems that a low vacuum is more effective, economically, with superheated than with moist

steam, although it undoubtedly is one of the chief considerations in both cases.

The following problem involves some considerations not

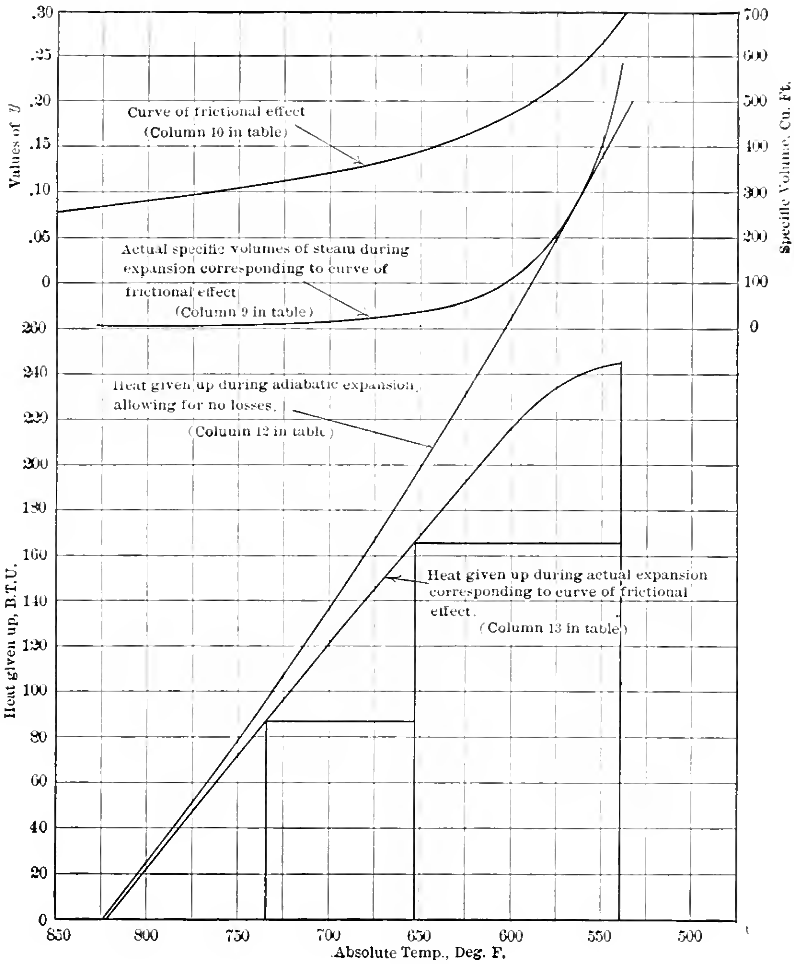



Fig. 77.

taken up in the foregoing examples, and the results correspond more nearly with the proportions adopted in practice. The curves and diagrams in Figs. 77 and 78 apply to this example.

*Example No. 6.*—Let a turbine be required to develop

1000 horse-power at full load, and be capable of using sufficient steam to produce 1500 horse-power, without the use of by-pass valves.

Let the initial pressure, at the throttle-valve, be 160 pounds absolute per square inch, and let the condenser maintain a vacuum of 29 inches of mercury. The upper and lower temperatures will then be 824° and 540° absolute respectively. The steam entering the turbine will be supposed to be dry and saturated.

Let the loss of energy due to friction of bearings, to exhaust velocity, and to windage be 18 per cent of the energy given up by the steam. The total heat given up by the steam, according to the curve in Fig. 77 is 245 B.T.U. per pound, and of this 82% is to be useful in developing energy of rotation. The steam consumption of the turbine will then be

$$\frac{1,980,000}{0.82\times245\times778}=12.7 \text{ pounds per delivered horse-power hour.}$$

When called upon for 50% overload the turbine will use more steam than this by, say, 16%, or the steam-channels must be so designed that they will accommodate 14.8 pounds per horse-power hour, when the machine is delivering 1500 horse-power. The steam used will then be

$$1500\times14.8=22,200 \text{ pounds per hour,}$$

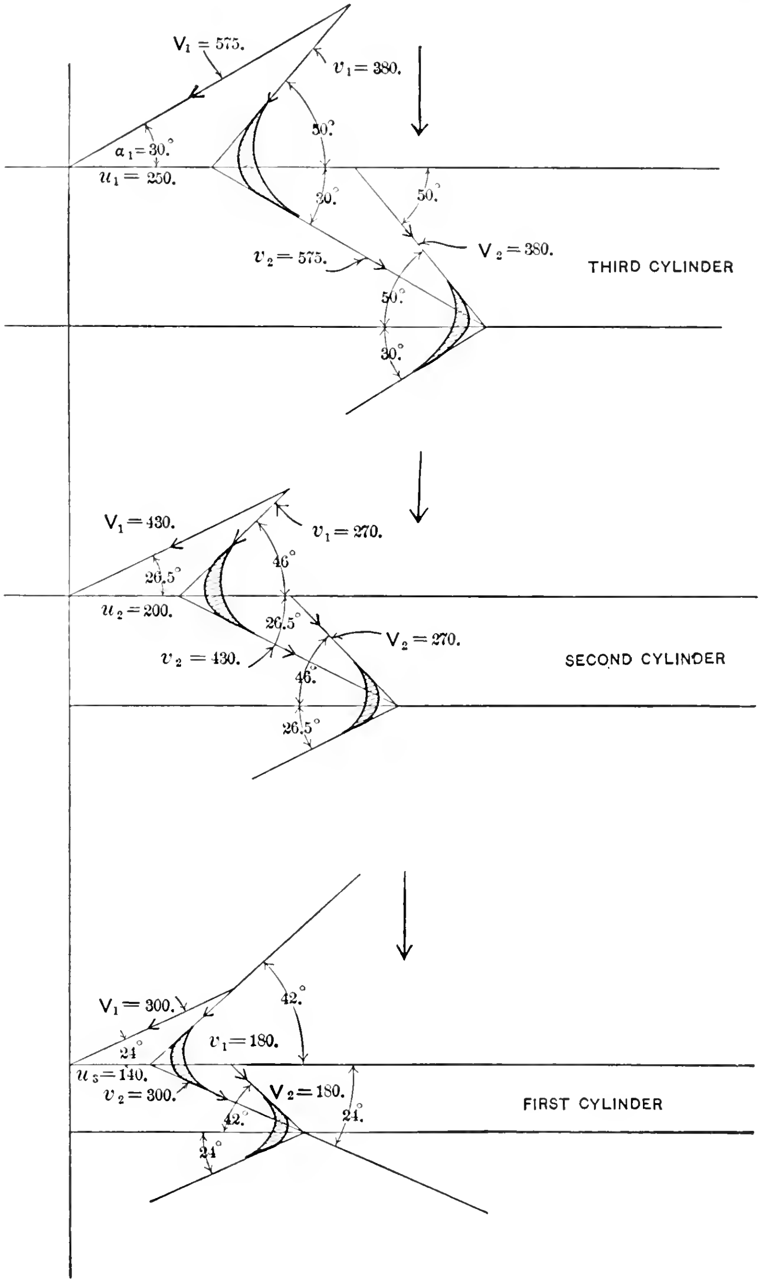
or

$$6.16 \text{ pounds per second.}$$

Let the pressure, temperature, and heat drop in the three cylinders be as follows:

| Pressure Absolute. | Temperature Drop. | Heat Drop. |
|---|---|---|
| 1. 160 to 45 pds. sq. in.. | 824−735= 89° | 86 B.T.U. |
| 2. 45 to 10 ins. mercury.. | 735−653= 82° | 80 " |
| 3. 10 ins. to 29 ins....... | 653−540=113° | 79 " |

Let the angles of exit of the moving blades equal those of the stationary blades, and have the values given below, for the various stages. These angles are varied to some extent from one set of blades to another, and in general are capable



V1 = 575.
v1 = 380.
50°
a1 = 30°
u1 = 250.
30°
50°
V2 = 380.
v2 = 575.
THIRD CYLINDER
50°
30°
V1 = 430.
v1 = 270.
46°
26.5°
u2 = 200.
26.5°
V2 = 270.
v2 = 430.
SECOND CYLINDER
46°
26.5°
42°
V1 = 300.
v1 = 180.
24°
u3 = 140.
v2 = 300.
V2 = 180.
24°
FIRST CYLINDER
42°
24°


FIG. 78.

of being "gaged" or set as may be found necessary for obtaining proper axial thrust conditions.

| Cylinder No. | Steam Velocity. | Peripheral Velocity. | Angle of Exit. |
|---|---|---|---|
| 1 | 300 | 140 | 24° |
| 2 | 430 | 200 | $26\frac{1}{2}$° |
| 3 | 575 | 250 | 30° |

The velocity diagrams for the three stages may now be drawn, as in Fig. 78, and the work done in each row of stationary and moving blades may be determined. Thus,

In first cylinder, work per row $=\frac{300^2-180^2}{64.4}=895$ foot-pounds, equivalent to 1.15 B.T.U.

Since there are 86 B.T.U. given up in this cylinder, and since the work in the moving rows is the same as that in the stationary rows, there will be $\frac{86}{2.3}=36$ moving and 36 stationary rows of blades in the first cylinder. In similar manner the number of blades in the second and third cylinders may be found with the following result:

| | |
|---|---|
| Moving blades—H.P. end of spindle | 36 |
| " " Middle cylinder on spindle | 18 |
| " " L.P. end of spindle | 14 |
| Total | 68 |

There will of course be an equal number of rows on the stationary casing of the machine.

**Diameter of Cylinders.**—Assuming the speed of revolution of the turbine as 2000 per minute, the mean diameters of the rows of blades in the various cylinders is fixed by the assumed mean peripheral ve'ocity of blades.

| Cyl. No. | Mean Diameter. |
|---|---|
| 1 | 1.335 feet or 16 inches. |
| 2 | 1.91 " " 23 " |
| 3 | 3.97 " " 47.5 " |

**Length of Blades.**—In each cylinder there are usually several barrels, each containing blades of a given length, but the blades of each barrel, advancing from the high-pressure end of the machine towards the condenser, are longer than those of the preceding barrel. In the first cylinder there may be three different lengths of blade, in the second four, and in the third five or six. The variation in length is for the purpose of increasing the cross-sectional area for the passage of steam as the latter expands in volume. The proper area for any section of the turbine may be calculated by finding the volume of the steam at that section from the curve of volumes, as plotted in Fig. 77.

The mean specific volumes of the mixture of steam and water while passing cylinders Nos. 1, 2, and 3, respectively, are, approximately, 4, 16, and 150 cu. ft. The weight of steam passing per second is 6.16 pounds, and the exit velocities are 300, 430, and 575 feet per second for the three cylinders respectively. Therefore the mean areas should be:

$$\text{1st cylinder, } \frac{6.16\times 4}{300} = 0.082 \text{ sq. ft.} = 9.9 \text{ sq. in.}$$

$$\text{2d} \quad \text{''} \quad \frac{6.16\times 16}{430} = 0.23 \quad \text{''} \quad \text{''} = 27.5 \quad \text{''} \quad \text{''}$$

$$\text{3d} \quad \text{''} \quad \frac{6.16\times 150}{575} = 1.60 \quad \text{''} \quad \text{''} = 1920 \quad \text{''} \quad \text{''}$$

Making the same assumption as to blade thickness as in the previous problem, the mean blade lengths for the three cylinders are

$$L_1 = \frac{1.5\times 9.9}{3.14\times 16\times \sin 24°} = 0.73'',\text{—say } \tfrac{3}{4}'';$$

$$L_2 = \frac{1.5\times 27.5}{3.14\times 23\times \sin 26.5°} = 1.18,\text{—say } 1\tfrac{3}{16}'';$$

$$L_3 = \frac{1.5\times 192}{3.14\times 47.5\times \sin 30°} = 3.87,\text{—say } 3\tfrac{7}{8}''.$$

**The thickness of the blades,** and the spacing employed, may be such that the blades occupy one-half or even two-thirds of the annular space between casing and spindle, and the factor allowing for this must be selected accordingly.

QUANTITIES USED IN CHARACTERISTIC CURVES IN FIG. 77.

| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
|---|---|---|---|---|---|---|---|
| $T_2$ | $(E_1-E_3)$ | $E_2$ | $H_v$ | $q_2$ | x′ | x″ | $x_2=x'+x''$ |
| 540 | 1.47 | 1.96 | 1058 | 47 | 0.750 | 0.0995 | 0.850 |
| 575 | 1.41 | 1.80 | 1034 | 82 | 0.783 | 0.064 | 0.847 |
| 600 | 1.36 | 1.70 | 1017 | 107 | 0.800 | 0.049 | 0.849 |
| 650 | 1.28 | 1.51 | 982 | 158 | 0.848 | 0.030 | 0.878 |
| 700 | 1.21 | 1.35 | 946 | 208 | 0.895 | 0.016 | 0.911 |
| 750 | 1.14 | 1.21 | 911 | 259 | 0.942 | 0.007 | 0.949 |
| 800 | 1.07 | 1.09 | 875 | 311 | 0.982 | 0.002 | 0.984 |
| 824 | 1.04 | 1.04 | 857 | 335 | 1.00 | ...... | ..... |

| 9 | 10 | 11 | 12 | 13 |
|---|---|---|---|---|
| $v=x_2$ (sp. vol. at $T_2$) | $y$ | $H_2=x'H_v+q_2$ | $H_1-H_2$ | $H=(H_1-H_2)(1-y)$ |
| 558 | 0.30 | 841 | 351 | 245 |
| 203 | 0.22 | 892 | 300 | 234 |
| 107 | 0.185 | 921 | 271 | 221 |
| 36 | 0.145 | 991 | 201 | 171 |
| 15 | 0.120 | 1055 | 137 | 121 |
| 7.0 | 0.100 | 1118 | 74 | 66 |
| 3.7 | 0.090 | 1160 | 32 | 20 |
| ... | 0.075 | 1192 | 0 | 0 |

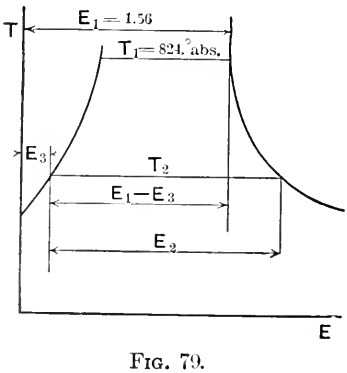



FIG. 79.

$H_v$ = heat of vaporization at $T_2$;
$q_2$ = heat of liquid at $T_2$;
x′ = quality after adiabatic expansion to $T_2$, $=\frac{E_1-E_3}{E_2}$;
x″ = increase of quality due to friction, $=y(H_1-H_2)\div H_v$;
$x_2$ = quality after actual expansion to $T_2$;
v = volume after actual expansion to $T_2$;
$y$ = energy loss;
$H_1$ = total heat in steam, per pd., at $T_1$, as it enters turbine;
$H_2$ = total heat in steam, per pd., after adiabatic expansion to $T_2$;
$H$ = heat given up during actual expansion to $T_2$.

**Comparison of Efficiencies of Single-stage Impulse- and Single-stage Reaction-turbines.**—The expressions for the hydraulic efficiency of the two types have been developed in preceding chapters and are as follows, for impulse- and for reaction-turbines respectively:

$$\text{Impulse-turbine,}\quad \text{Efficiency} = \frac{4u}{V_1}\left\{\cos\alpha - \frac{u}{V_1}\right\};$$

$$\text{Reaction-turbine,}\quad \text{Efficiency} = \frac{u}{V_1}\left\{2\cos\alpha - \frac{u}{V_1}\right\}.$$

These equations are plotted on Plates XIV and XV, and the variation of maximum efficiency with $\frac{u}{V_1}$ and with the angle at which the steam leaves the nozzles or the guide-blades is shown on Plate XVI.

Expressed numerically, the curves on Plate XIV show the following values of the ratio $\frac{u}{V_1}$ for the conditions stated:

| | | $\frac{u}{V}$ for Max. Efficiency. |
|---|---|---|
| Impulse-turbine | $\alpha = 10°$ | .49 |
| | 20° | .48 |
| | 30° | .44 |
| | 40° | .38 |
| Reaction-turbine | $\alpha = 10°$ | .97 |
| | 20° | .93 |
| | 30° | .87 |
| | 40° | .80 |

The steam velocities used in the impulse-turbine are much higher than in the reaction-turbine, but the ratios of peripheral to steam velocity, for maximum efficiency, are lower. In the compound impulse-turbine the work done in each stage is greater than that done in the reaction-turbine per stage, and there are therefore fewer stages in the impulse-turbine.

In the impulse-turbine the efficiency is zero when $\cos \alpha = 1$ and $u = V$; that is, when the jet follows directly behind the buckets, with the same velocity as the buckets.

Plate XVII shows the variation of efficiency for the compound impulse-turbine, with $\alpha$ and $u$, for varying number of stages.

In the reaction-turbine the efficiency is zero when $\cos \alpha = 1$, and $u = 2V$.

While the reaction-turbine requires a greater value of the ratio $\frac{u}{V}$ for maximum efficiency than does the impulse-turbine, its greater number of stages causes the steam velocity produced per stage to be much lower. This permits the attainment of satisfactory efficiency at comparatively low peripheral velocities. The following particulars applying to Parsons turbines are from a paper by Mr. E. M. Speakman.*

ELECTRICAL WORK.

| Normal Output of Turbine. | Peripheral Vane Speed. | | Number of Rows. | Revolutions per Minute. |
|---|---|---|---|---|
| | First Expansion. | Last Expansion. | | |
| 5000 K.W. | 135 | 330 | 70 | 750 |
| 3500 K.W. | 138 | 280 | 75 | 1200 |
| 2500 K.W. | 125 | 300 | 84 | 1360 |
| 1500 K.W. | 125 | 360 | 72 | 1500 |
| 1000 K.W. | 125 | 250 | 80 | 1800 |
| 750 K.W. | 125 | 260 | 77 | 2000 |
| 500 K.W. | 120 | 285 | 60 | 3000 |
| 250 K.W. | 100 | 210 | 72 | 3000 |
| 75 K.W. | 100 | 200 | 48 | 4000 |

MARINE WORK.

| Type of Vessel. | Peripheral Vane Speed. | | Mean Ratio, $u \div V$. | Number of Shafts. |
|---|---|---|---|---|
| | H.P. | L.P. | | |
| High-speed mail steamers | 70– 80 | 110–130 | 0.45–0.5 | 4 |
| Intermediate do. | 80– 90 | 110–135 | 0.47–0.5 | 3 or 4 |
| Channel steamers | 90–105 | 120–150 | 0.37–0.47 | 3 |
| Battle-ships and large cruisers | 85–100 | 115–135 | 0.48–0.52 | 4 |
| Small cruisers | 105–120 | 130–160 | 0.47–0.5 | 3 or 4 |
| Torpedo craft | 110–130 | 160–210 | 0.47–0.51 | 3 or 4 |

* Trans. Inst. of Engineers and Shipbuilders of Scotland, 1905–6.

PLATE XIV.

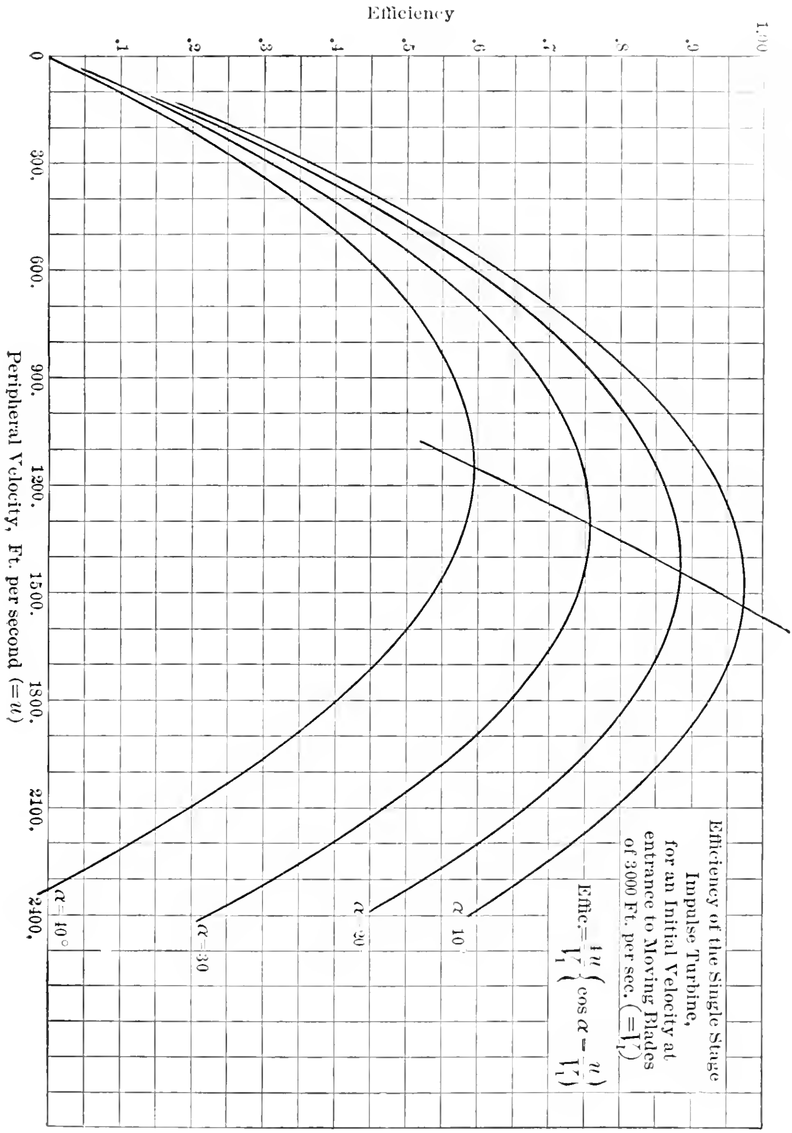
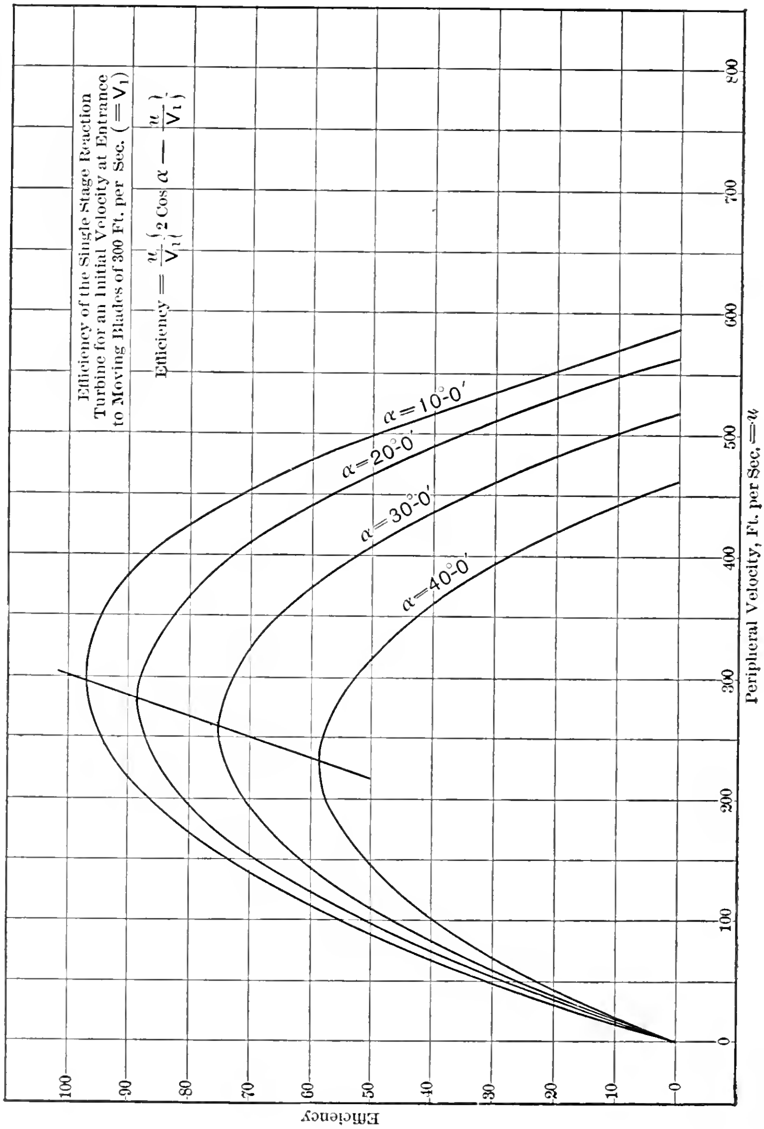

Efficiency
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.00
300.
600.
900.
1200.
1500.
1800.
2100.
2400.
Peripheral Velocity, Ft. per second (=u)
Efficiency of the Single Stage Impulse Turbine, for an Initial Velocity at entrance to Moving Blades of 3000 Ft. per sec. (=$V_1$)
Effic.= $\frac{4u}{V_1}\left(\cos\alpha - \frac{u}{V_1}\right)$
$\alpha = 10°$
$\alpha = 20°$
$\alpha = 30$
$\alpha = 40°$

PLATE XV


Efficiency of the Single Stage Reaction Turbine for an Initial Velocity at Entrance to Moving Blades of 300 Ft. per Sec. (= $V_1$)
Efficiency = $\frac{u}{V_1}\left\{2 \cos \alpha - \frac{u}{V_1}\right\}$
$\alpha = 10°\text{-}0'$
$\alpha = 20°\text{-}0'$
$\alpha = 30°\text{-}0'$
$\alpha = 40°\text{-}0'$
Efficiency
100
90
80
70
60
50
40
30
20
10
0
Peripheral Velocity, Ft. per Sec. = $u$
0
100
200
300
400
500
600
700
800

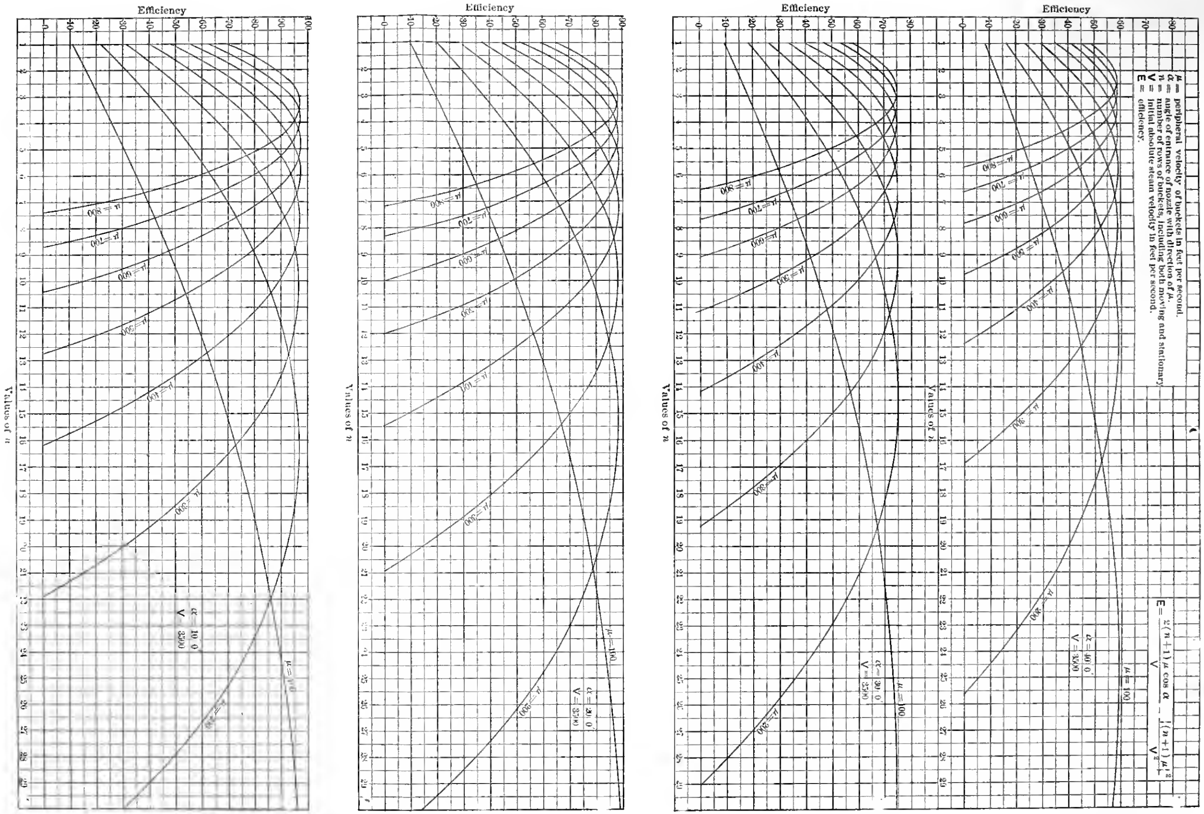
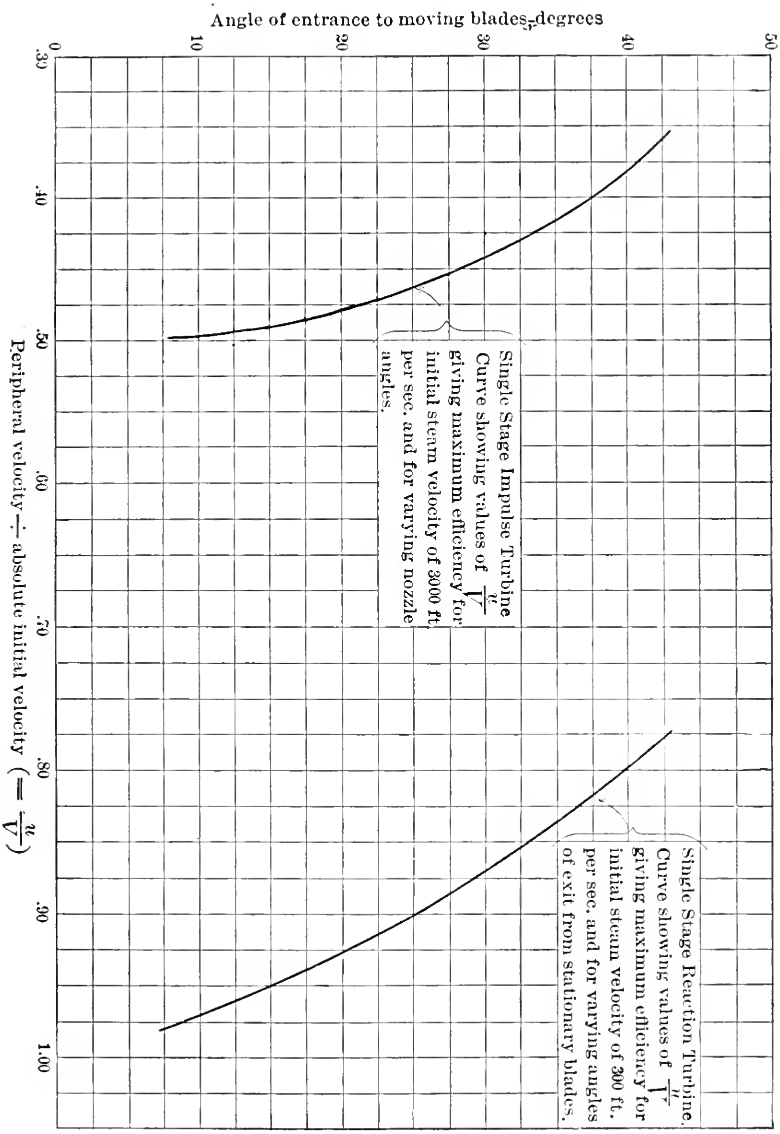

Efficiency
Values of n
α = 10° 0'
V = 3500
α = 30° 0'
V = 3500
α = 40° 0'
V = 3500
μ = 100
μ = 200
μ = 300
μ = 400
μ = 500
μ = 600
μ = 700
μ = 800
μ = peripheral velocity of buckets in feet per second.
α = angle of entrance of nozzle with direction of μ.
n = number of rows of buckets, including both moving and stationary.
V = initial absolute steam velocity in feet per second.
E = efficiency.
E = 2(n+1)μ cos α / V − [(n+1)μ]² / V²

PLATE XVI.


Angle of entrance to moving blades,-degrees
0
10
20
30
40
50
.30
.40
.50
.60
.70
.80
.90
1.00
Peripheral velocity ÷ absolute initial velocity $\left(= \frac{u}{V}\right)$
Single Stage Impulse Turbine Curve showing values of $\frac{u}{V}$ giving maximum efficiency for initial steam velocity of 3000 ft. per sec. and for varying nozzle angles.
Single Stage Reaction Turbine. Curve showing values of $\frac{u}{V}$ giving maximum efficiency for initial steam velocity of 300 ft. per sec. and for varying angles of exit from stationary blades.

## Heat Analysis of Steam Turbines.

In as much as the calculations by means of which the steam passages are proportioned presuppose some law of expansion of the steam in the turbine, it is very desirable that such tests of completed turbines should be carried out as would show to what extent the performance actually attained agrees with that assumed as the basis of calculation by the designer.

An analysis of a turbine, based upon heat expenditure, should show what percentage of the available heat in the steam is given up in each section of the turbine. This would involve measurement of average temperatures and pressures where superheat exists, and average pressures and qualities where the steam is not superheated. The condition of the steam within a turbine in operation is far from uniform over any given cross-section of the steam path, and it is therefore necessary to take temperatures and qualities at many different depths in any steam passage, in order to obtain average results as to the heat contents of the steam.

The information required for an analysis includes:

(*a*) Horse-power delivered from the turbine shaft.

(*b*) Steam used per unit of time.

(*c*) Average pressures, temperatures and qualities at certain points along the turbine, including the last section.

(*d*) The weight of steam collected as drainage-water, if any, from the various parts of the turbine.

Assuming that determinations of quality at the various nozzle bowls of impulse turbines, and between the sections of Parsons turbines can be satisfactorily made, the amount of heat given up by the steam in each stage, or section, may be determined. The amount of water drained off from each stage should be measured, especially in the case of impulse turbines, and the heat so carried away be allowed for. A curve of expansion of the steam may then be drawn on a heat diagram (see chart on back cover of book), from which the steam consumption may be computed for a turbine supposed to have no radiation and

bearing friction losses. By comparisons of this steam consumption with that actually determined, per B.H.P., by test, the factor may be found by means of which actual steam consumption may be predicted from the curve of heat given up, as drawn on the heat-diagram.

Suppose that the curve on the temperature-entropy chart shows that each pound of steam gives up 185 B.T.U. during its passage through the turbine. Then the water-rate, not consid-

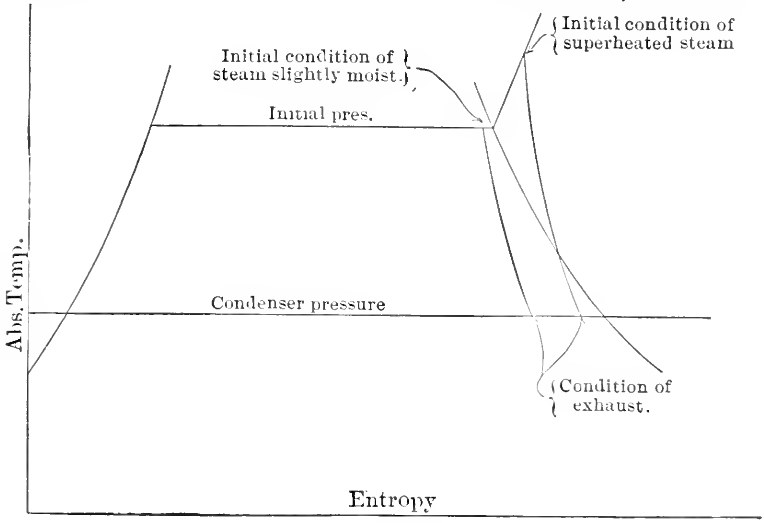



FIG. 80.

ering the external losses of radiation and mechanical friction would be

$$\frac{1,980,000}{778 \times 185} = 13.8 \text{ pounds per H.P.-hour.}$$

Suppose the water-rate is found from the test to be 16 pounds per brake horse-power-hour. This means a useful expenditure of

$$\frac{1,980,000}{778 \times 16} = 159 \text{ B.T.U. per pound of steam.}$$

Therefore the mechanical-friction and radiation losses use up $185 - 159 = 26$ B.T.U. for each pound of steam passing through the turbine.

**Results to be expected from a given design** may be predicted

by means of expansion curves obtained from tests of similar turbines as follows:

The initial pressure and superheat or quality to be used having been determined, and the vacuum being assumed, the final condition of the steam may be calculated, corresponding to a given water-rate.

Thus, if the water-rate is to be 14 pounds per B.H.P. hour, the heat usefully employed per pound of steam will be

$$\frac{1,980,000}{14 \times 778} = 182 \text{ B.T.U.}$$

Besides this expenditure of heat, the surrounding air has been heated by radiation from the turbine, and also the surrounding air and the oil and water used in the bearings have been heated by the heat appearing as bearing friction. Also, the water drained from the turbine has carried away some heat. These quantities of heat have not been returned to the steam as is the case with the heat of friction between the steam itself and the passageways in the turbine. The quality in the exhaust pipe will, therefore, represent a smaller final heat contents than that obtained by subtracting 182 B.T.U. from the total heat in the entering steam.

Let $H_1$ = heat in entering steam per pound.
$H_W$ = heat appearing as B.H.P. per pound.
$H_L$ = heat appearing as the losses mentioned above.
$H_2$ = heat in exhaust steam per pound.

Then $H_2 = H_1 - (H_W + H_L)$

Since $H_2$, $H_1$ and $H_W$ are all known or capable of determination, the amount of the losses, $H_L$, may be found.

Thus, if $H_1$ = 1250 B.T.U. per pound,
$H_W$ = 182 B.T.U. per pound,
$H_2$ = 1050 B.T.U. as found by calorimeter determinations,

Then, $H_L = H_1 - H_2 - H_W = 1250 - 1050 - 182 = 18$ B.T.U.

By means of the values of $H_L$ as found from analyzing turbine tests in the above manner, the conditions of expansion in a proposed similar design may be predicted and the proportions of the nozzles, steam passages, etc., for a given energy distribution may be calculated.

*Example.*—Suppose a turbine receives steam at 175 pounds abs. and 160 deg. F. superheat, and that the vacuum is 28″ mercury.

From the heat diagram the initial heat contents 1298 B.T.U. Suppose calorimeter determinations show the average quality in the exhaust from the last set of buckets or blades to be .91. The heat contents of the exhaust as found from the heat diagram will be $H_2' = 1020$ B.T.U., approximately. Let the water rate be found by test to be 11.5 pounds per brake horse-power-hour.

If there were no losses due to mechanical friction, radiation, leakage, etc., the water rate would be

$$\frac{1,980,000}{778 \times (H_1 - H_2)} = \frac{2545}{1298 - 1020} = \frac{2545}{278} = 9.51 \text{ pounds per hour.}$$

Due to the losses, $H_L$, the water rate is raised to 11.5 pounds, or

$$\frac{2545}{278 - H_L} = 11.5.$$

Therefore, the external losses amount to

$$H_L = 278 - \frac{2545}{11.5} = 57 \text{ B.T.U.}$$

If the steam had expanded adiabatically the final heat contents $H_2$, would have been (from the heat diagram) 930 B.T.U. per pound, and the steam consumption

$$\frac{2545}{1298 - 930} = 6.91 \text{ pounds.}$$

The efficiency of the turbine is therefore $\dfrac{6.91}{11.5} = .60.$

A steam calorimeter has recently been brought out by the author, by means of which the quality may be determined of any steam which can be caused to pass through the instrument. The instrument is applicable in the case of the lowest as well as the highest pressures used in practice, and the degree of wetness of the steam may have any value whatever without affecting the accuracy of the determinations. The problem, however, of obtaining representative samples of steam presents the most serious obstacles at present in the way of thermal analysis of steam turbines.

**Amount of Superheat to be Used in Turbines.**—It is desirable that the degree of superheat be as high, but not higher, than that which will prevent moisture from being produced before the steam has passed through the last stage. This is because of the following:

(*a*) Moisture in the steam is supposed to cause losses due to friction between steam and buckets, and to increase rotation losses.

(*b*) If the degree of superheat is great enough so the exhaust

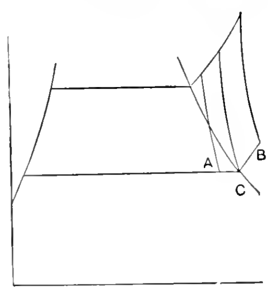



FIG. 81.

is superheated, the superheat carried away by the exhaust is lost.

(*c*) The maintenance of superheaters and of the machinery in general is more expensive the higher the superheat.

If the expansion curve, as determined from the temperatures and qualities in the various stages of actually tested turbines ends at *A*, Fig. 81, it is evident that the degree of

superheat is too low and that some of the stages are working in wet steam with the consequent losses.

If the expansion curve ends at *B*, there is superheat in the exhaust, and resulting loss of efficiency caused by too high initial superheat.

A curve *C*, representing the correct condition of the exhaust, may be drawn on the diagram, similar in character to the curve of expansion already determined, and indicating approximately the degree of superheat which will be necessary in order that the exhaust may be just dry and saturated.

**Description of the Calorimeter.**—Figures 82 and 83 show the exterior and the general interior arrangement of a steam calorimeter with which the quality of any steam passing through the calorimeter can be determined with accuracy in a very simple manner. The instrument is especially designed for determining the quality of steam at different points along steam turbines, and it can be used with steam of any degree of wetness and of any temperature and pressure above that in the condenser. The sampling-tube leading to the calorimeter, and shown in Fig. 86, may be extended into any steam passage from one part of the turbine to another, and the average quality may thus be investigated by taking successive samples from different depths. From this information, combined with the results of ordinary tests, a curve may be drawn on a heat diagram, showing the distribution of the work done by the steam in the turbine, and indicating the efficiency of the various sets of blades or buckets. Such a curve may also be used to indicate the degree of initial superheat that should be used in the entering steam in order that the steam at any set of blades may be in a given desired condition as to heat contents.

The difficulty of obtaining a representative sample of steam, especially under the complex conditions existing in steam turbines, is fully recognized, and the sampling-tube shown in Figs. 85 and 86 has been designed to assist in obtaining definite results. With this tube it is at least possible to obtain samples from given definitely known depths or positions in a steam passage.

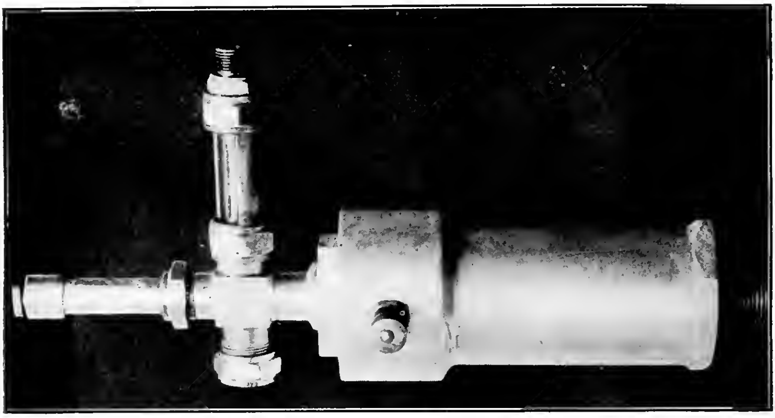


Fig. 82.

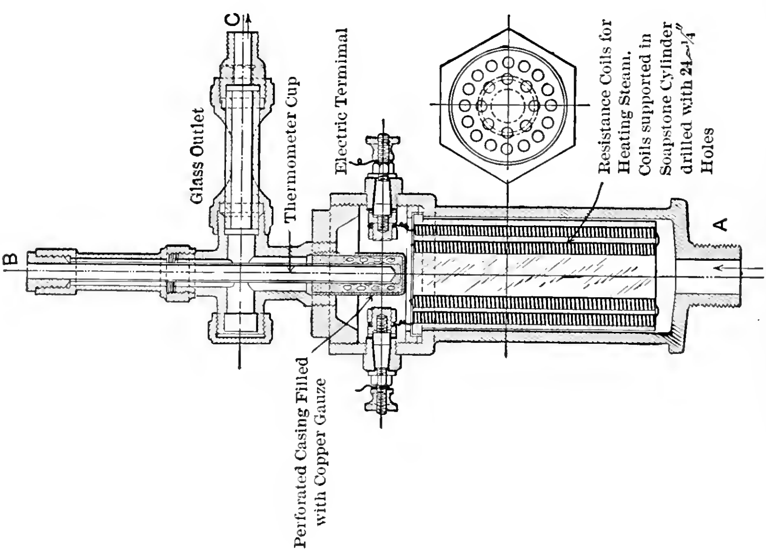

B
C
Glass Outlet
Thermometer Cup
Electric Terminal
Resistance Coils for Heating Steam. Coils supported in Soapstone Cylinder drilled with 24–1/4″ Holes
A
Perforated Casing Filled with Copper Gauze

Fig. 83.

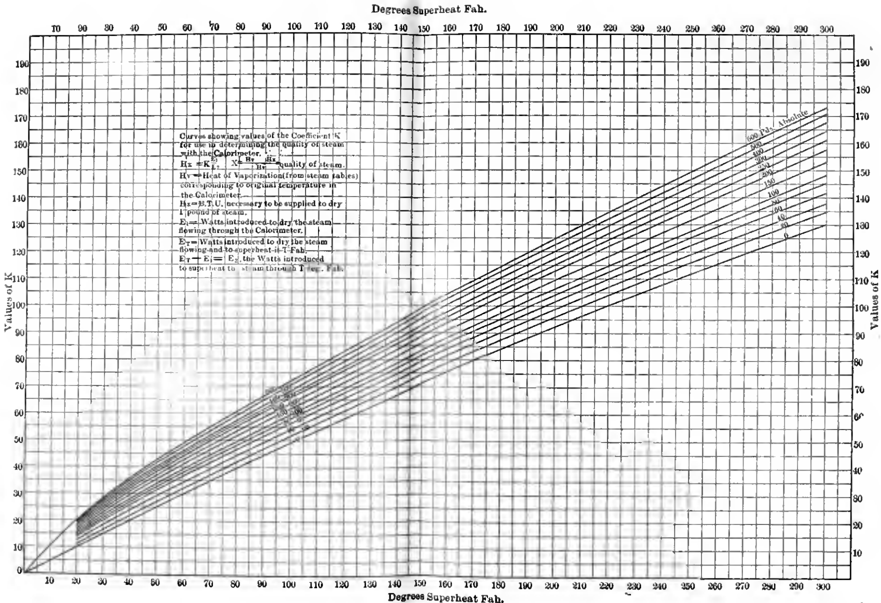



Fig. 82.

*To face page 236.*

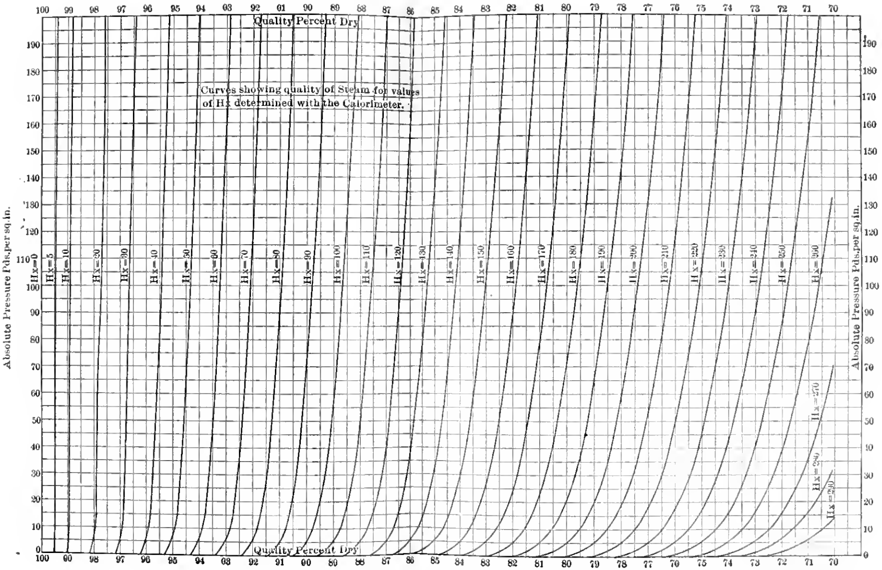



FIG. 83

*To face page 236.*

The development of the instrument resulted from experiments in passing steam from an electrically heated calorimeter through a transparent glass tube. If the electrical energy supplied to the steam in passing through the calorimeter was insufficient to dry the steam, water could be seen in the interior of the glass tube; also no rise of temperature was shown on a thermometer placed in the tube. By adding sufficient electrical energy to the steam, the interior of the tube cleared up because of the disappearance of the water, and the temperature began to rise at the same instant that the steam gave this evidence of being completely dry. It was therefore possible to measure directly the amount of heat required to dry the sample of steam, and from the known heat of vaporization of dry steam, and the known weight of steam passing through the calorimeter, the quality could be readily found. The method of ascertaining the weight of steam passing per unit of time is described in the following paragraph.

In operation the calorimeter is attached to the sampling-tube, or to the source of steam directly, by means of the screw-thread $A$. Steam is thus admitted to the instrument, from which it passes to the condenser or to the atmosphere, through a pipe from the discharge valve, placed at $C$. Having adjusted the discharge valve so it is passing a suitable quantity of steam, enough energy is turned in to heat the steam to dryness. The condition of dryness is indicated by an immediate rise of temperature as shown by the thermometer, if more than the requisite amount of electrical energy is supplied. For convenience let the watts necessary to dry the steam be called $E_1$. After noting this number of watts the steam is further heated by additional watts $E_2$, till a temperature $T$ degrees above saturation is obtained, say 20, 30, 100, or some other convenient number of degrees superheat. This operation is for the purpose of ascertaining the weight of dry steam per hour, $W_1$, which was passing when the steam was just dry. The weight $W_2$, passed through the valve after superheating, will be less than the dry steam passed through, by some percentage represented by a

constant, $C$, because of the increase in volume accompanying superheating. This has been determined by test with the instruments. Also, the watts $S$, necessary to superheat a pound of steam to $T$ degrees, at different pressures, have been determined. The latter, $S$, may be used instead of the specific heat of steam and has the advantage of allowing automatically for radiation losses. The quality of the steam may be obtained either by use of the curves supplied with the calorimeter, or by the use of the specific heat of steam for varying pressures. The use of the curves, however, eliminates entirely possible errors due to uncertainty as to the value of the specific heat.

Let $W_1$ = weight of dry steam passing per hour.

Then $E_1$ watts evaporates the moisture in $W_1$ pounds per hour and this energy is equivalent to $\dfrac{3.412E_1}{W_1}$ thermal units per pound of steam. Let this amount of heat be represented by $H_x$. Let $W_2$ = weight of superheated steam passing per hour $= CW_1$. Then $E_2$ watts raises the temperature of $CW_1$ pounds of steam through $T$ degrees, or $E_2 = CW_1S$ watts, where $S$ represents the watts required to superheat one pound of steam per hour through $T$ degrees, at the pressure in the calorimeter.

Then, $$W_1 = \frac{E_2}{CS},$$

and this value of $W_1$ may be substituted in the equation,

$$H_x = \frac{3.412E_1}{W_1}.$$

Thus, $$H = \frac{3.412E_1 \times CS}{E_2}.$$

Since $C$ and $S$ are constants for any given pressure and degree of superheat, 3.412 $CS$ may be written as a constant, $K$, and values of this constant are given for varying pressures and de-

grees of superheat, by a set of curves plotted from experimentally obtained data. See Fig. 82.

The equation then becomes

$$H_x = K\frac{E_1}{E_2}.$$

If $H$ represents the heat of vaporization of dry steam (from steam tables) at the pressure indicated by the original temperature in the calorimeter, the quality of the steam passing through the calorimeter is

$$x = \frac{H_v - H_x}{H_v}.$$

It is to be noted that the constant $K$ is independent of the weight of steam flowing through the calorimeter during superheating, although consideration of this weight has been included in the explanation just given. The same result may be found without the use of either $C$ or $S$. The independence of $K$ upon these quantities may be shown as follows:

As defined above,

$$C = \frac{W_2}{W_1}, \quad \text{and} \quad S = \frac{E_2}{W_2}.$$

$$K = 3.412SC = 3.412 \times \frac{E_2}{W_2} \times \frac{W_2}{W_1} = \frac{3.412E_2}{W_1}.$$

It is therefore possible to find the expression for $K$ without using $C$ or $S$. Thus, if the steam flowing through the calorimeter is first dried, by the introduction of $E_1$ watts, then superheated through $T$ degrees by a further introduction of $E_2$ watts, it follows that for each pound of the original weight $W_1$ of dry steam, it has taken $\frac{E_2}{W_1}$ watts to superheat the steam coming from $W_1$ pounds of dry steam, through the given temperature range $T$. Hence, for the pressure and temperature in question, $\frac{E_2}{W_1}$ is a constant, which may be called $k$, and $W_1 = \frac{E_2}{k}$.

Substituting this value of $W_1$ in the equation $H_x=\frac{3.412E_1}{W_1}$,

$$H_x=\frac{3.412E_1k}{E_2}.$$

Let $3.412k$ be called $K$.

Then $$H_x=K\frac{E_1}{E_2}, \text{ as before found.}$$

Summing up the operations involved in determining the quality of steam, they are as follows:

1. The calorimeter is attached to the source of steam and a flow of steam takes place through the electrical heating coils, and out through the discharge valve at $C$, Fig 83. Electrical connections are made with an ordinary D. C. circuit at perhaps 125 volts, capable of carrying 10 amperes. The calorimeter is put in series with the water rheostat, and means are provided for measuring the input of electrical energy.

2. Electrical energy is supplied sufficient not only to dry the steam, but to superheat it to some convenient temperature. The watts introduced are called $E_T$. Now turn out energy until superheating no longer takes place, and the steam is therefore just dry. The energy now being introduced is only that necessary to dry the steam, and is called $E_1$. The condition of dryness is indicated as before described. Then $E_T-E_1=E_2$ watts required to superheat through $T$ deg.

3. From the curves select the value of the coefficient $K$ corresponding to the degree to which the steam was superheated, and to the original temperature in the calorimeter, and find the heat,

* The constant 3.412 is the number of B.T.U. equivalent to 1 watt-hour. Its development may be shown as follows:

1 horse-power hour is equivalent to $33000\times60$ or 1,980,000 foot-pounds.
1 B.T.U. is equivalent to 778 foot-pounds.

Therefore, 1 horse-power hour is equivalent to $\frac{1,980,000}{778}-2545$ B.T.U,

But 1 horse-power hour is also equivalent to 746 watts.

Therefore, 1 watt-hour is equivalent to $\frac{2545}{746}=3.412$ B.T.U.

$H_x$, which has been added to each pound of steam in order to dry it, from the equation $H_x = K\frac{E_1}{E_2}$.

4. Find the quality of steam from the second set of curves, Fig. 83, representing the equation $x = \frac{H_r - H_x}{H_r}$. The quality may of course be found directly from the equation if desired.

The question may be asked, why not call the watts required to dry and superheat the steam $E_1 + E_2$, instead of $E_T$. It will be seen upon reflection that when drying and superheating are taking place together, $E_1$, as prevoiusly defined, is not being introduced to *dry* the steam, because the steam passing through the orifice or valve at outlet from the calorimeter is less during superheating than during drying of the steam, and therefore the heat necessary to dry the steam passing through the calorimeter is less then $E_1$. As soon as $E_2$ has been turned out, and the steam is just dry, $E_1$ is again being introduced, but $E_1$ and $E_2$, as defined, are not simultaneously introduced.

*Example No.* 1.—Let wet steam be passing through the calorimeter at a temperature of 366 deg. F., as shown by the thermometer in the tube $B$. This corresponds to a pressure of 165 pounds per square in. abs. Let 650 watts $(=E_T)$ be introduced, and let the resulting temperature be 460 deg. The steam has then been not only dried, but superheated through a range of 94 deg. $(=T)$. Let the energy be now reduced until the steam is just dry, and let the watts then being introduced be 260 $(=E_1)$.

$$E_2 = E_T - E_1 = 650 - 260 = 390.$$

From the curves the value of $K$ corresponding to this pressure and to $T = 94$ deg., is $K = 59.0$.

Therefore, $$H_x = \frac{260}{390} \times 59.0 = 39.4,$$

and from the curves of quality, $x = 95.4$ per cent.

*Example No.* 2.—Let the pressure as indicated by the temperature of 153 deg. in the calorimeter be 4 pounds abs.

Let $E_T=480$ watts, and let the resulting temperature $=240$ deg., corresponding to a range of 87 deg. superheat.

Let $E_1$ be found to equal 360 watts. Then $E_2=480-360=120$.

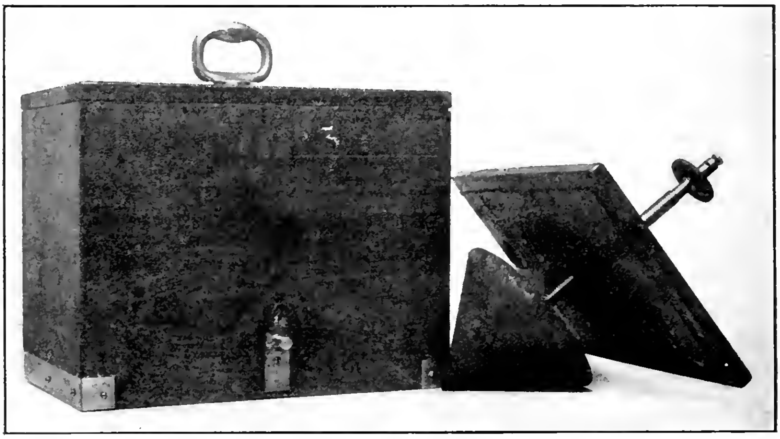
From the curves, $K=43.5$,

$$H_x=\frac{320}{120}\times 43.5=116.$$

From the curves of quality,

$$x=88.7 \text{ per cent.}$$

The calorimeter is supplied with a water rheostat box which serves to control the amount of electrical energy introduced, and also for carrying the calorimeter and accessories. The complete outfit is shown in Fig. 84 together with a separate cover



*A* *B*

*A*—Calorimeter Outfit Complete.
*B*—Cover of Water-rheostat Box, Fitted with Adjustable Terminal.

FIG. 84.

for showing how the rheostat is operated. When the box is in use as a water rheostat the cover holds a nut in which a screw, *D*, works for raising or lowering the cone *E*, and thus varying the energy passing the rheostat. The lower terminal connection is

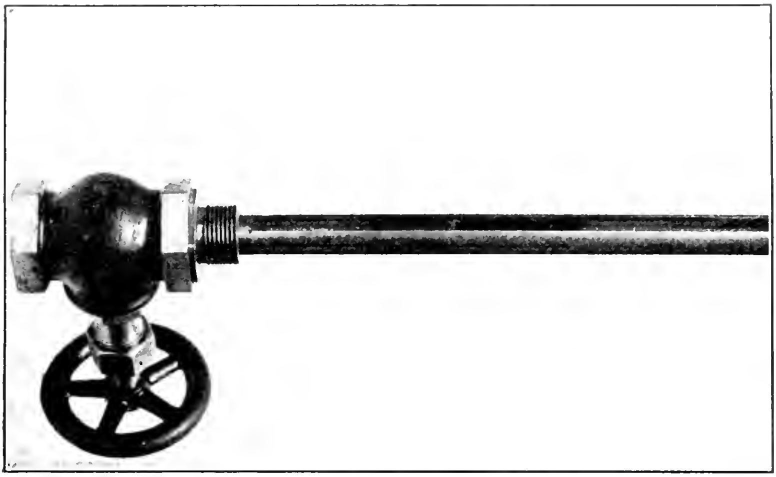


FIG. 85.

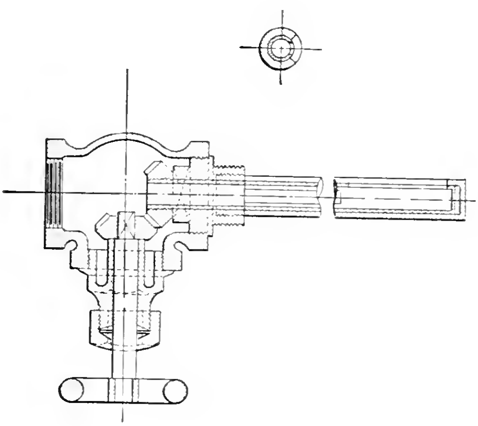


FIG. 86.

shown at *F*, on the box, and the upper is at *G*, in the swivel nut on top of the operating screw. The box is about 13 inches high, and the outfit complete weighs about 43 pounds. When the calorimeter is in the box it is attached to the iron plate forming the lower terminal of the rheostat on the bottom of the box, by means of the screw thread *A*, and the top of the calorimeter projects through the top of the box one-half inch. The brass handle is tapped out and screws on the top of the calorimeter for convenience in carrying the outfit.

The cover shown at the right of Fig. 84 belongs to another rheostat, not to the complete outfit shown at the left of the figure. The box to the left shows the calorimeter outfit ready for transportation. The terminal shown on the outside of the box can be unscrewed and carried with the other accessories inside the box.

The glass outlet from the calorimeter is not a necessary part of the instrument, since the condition of dryness is indicated by the rise of the mercury in the thermometer, but it is useful as giving an optical demonstration that the steam has been completely dried, and also in affording an excellent means for studying the behavior of wet and of superheated steam.

The sampling-tube shown in Figs. 85 and 86 permits of taking consecutive samples of steam from different depths in any steam passage. There are two tubes, of which the outer is stationary, and the inner can be rotated by means of the hand-wheel and bevel-gear connections, The outer tube is slotted over its entire length, while the inner tube contains short slots which open consecutively into the long slot in the outer tube. It is thus possible to take samples from the different portions of a steam passage without disconnecting the calorimeter, and to know positively from what part the sample is being drawn.

# CHAPTER IX.

## TYPES OF TURBINE AND THEIR OPERATION.

Detailed descriptions of the steam-turbines in use at the present time are available in catalogues, and in technical books and papers, so that in the following discussion only the dis-

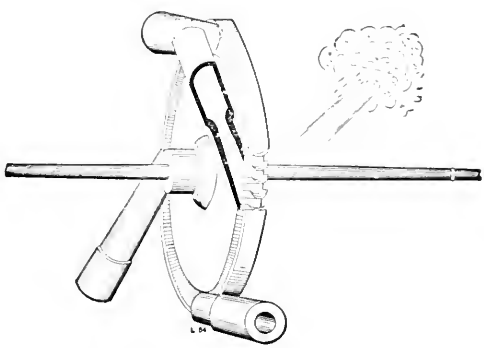


Fig. 87.—Simple impulse-wheel, De Laval type.

tinctive features of certain representative types will be dealt with.

The turbines that have been developed commercially in this country are of three types: (*a*) the De Laval; (*b*) the Parsons; (*c*) the Curtis.

**The De Laval Turbine** is shown in Figs. 87 to 91. It is a simple impulse-turbine, consisting essentially of a single wheel or disk, upon the rim of which are mounted buckets, or blades, which receive impulse from a set of expanding nozzles delivering steam at high velocity. The buckets are placed radially around

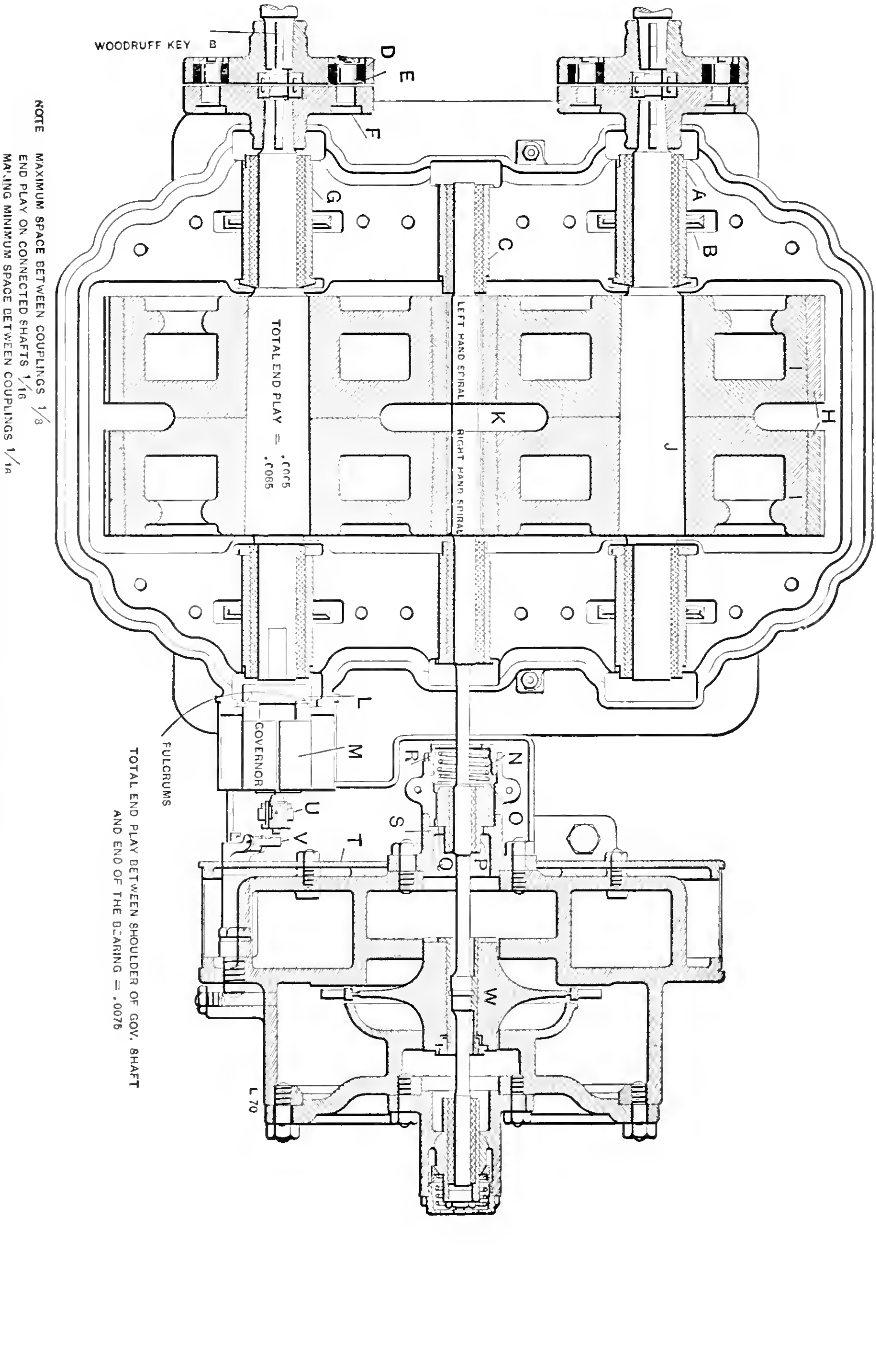



FIG. 88.—De Laval steam-turbine and spiral gearing.

the circumference of the wheel, and the nozzles are distributed about the circumference as shown in Fig. 87.

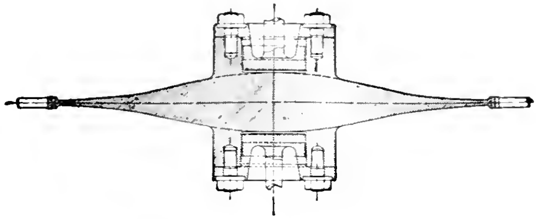


FIG. 89 —De Laval turbine rotor.

The essential parts of this turbine are:

(*a*) The nozzles, in which the steam expands to the condenser pressure, and attains the maximum possible velocity under the conditions of operation.

(*b*) The blades, or buckets, which change the direction of

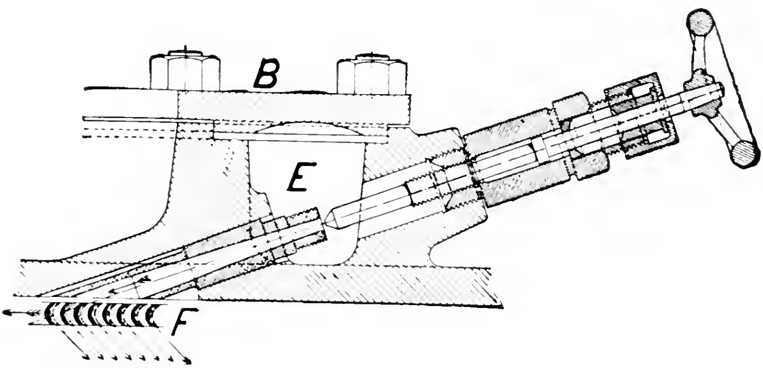



FIG. 90 —De Laval nozzle and valve.

the flow of steam, and thereby transform the energy of the jet into useful work in turning the shaft upon which the wheel is mounted.

A distinguishing feature of this type of turbine is the high speed of rotation of the wheel. This is made necessary because, in order efficiently to utilize the energy of the steam-jet, the peripheral velocity of the buckets must be from 0.35 to 0.5 of the velocity of the steam leaving the nozzles. The high peripheral velocity could be obtained at a low speed of revolution

if the wheel diameter were to be made correspondingly large. But large diameters are impracticable because of the frictional forces which would be brought into play, and certain proportions have been found which permit of a reasonable peripheral speed and allowable stresses in material of the wheel or disk. The speed of revolution, however, remains high, and can be utilized for driving machinery only by the use of gearing. The number of revolutions per minute varies from 8000 or

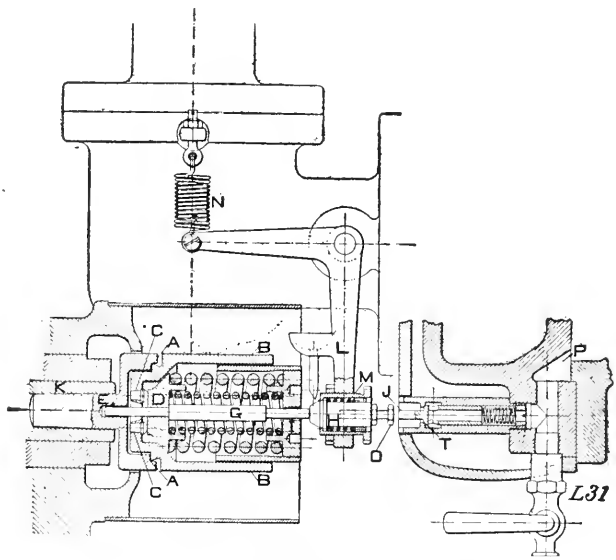



FIG. 91.—De Laval governing mechanism.

10,000, in 300-horse-power turbines, to 25,000 or 30,000, in very small machines. Since it is impracticable perfectly to balance a wheel of the type used, a light, flexible shaft is employed, which allows the wheel to assume its proper center of rotation, and thus to operate like a truly balanced rotating body.

The De Laval turbine has the advantage of developing a large amount of power per unit of weight, and is readily applied to the driving of electric generators, centrifugal pumps, and blowers.

DE LAVAL 300-HORSE-POWER TURBINE-TESTS, CONDUCTED BY MESSRS. DEAN AND MAIN.

TESTS WITH SATURATED STEAM.

Number of nozzles open, eight (8).
Average reading of barometer, 29.92 in.
Average temperature of room, 90° F.

| Date, 1902. | Hour. | Feed-water Weighed per Hour, Lbs. | Condensation from Separator, Lbs. | Moisture in Steam at Throttle by Calorimeter. | Dry Steam Entering Turbine, Lbs. | Pressure above Governor-valve, Lbs. | Pressure below Governor-valve, Lbs. | Vacuum, In. | R.P.M. of Generators. | Brake Horse-power. | Dry Steam used per Brake H.P. per Hour, Lbs. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| May 23<br>May 23 | 8.15 A.M.<br>9.15 A.M. | 5289 | 70 | 2.15% | 5107 | 204.7 | 196.2 | 26.7 | .... | 332.2 | 15.37 |
| May 23<br>May 23 | 9.15 A.M.<br>10.15 A.M. | 5073 | 70 | 2.15% | 4896 | 206.2 | 196.2 | 26.6 | .. | 332.4 | 14.73 |
| May 23<br>May 23 | 10.15 A.M.<br>11.15 A.M. | 5286 | 70 | 2.15% | 5104 | 207.2 | 196.3 | 26.6 | ... | 332.2 | 15.37 |
| May 23<br>May 23 | 11.15 A.M.<br>12.15 A.M. | 5283 | 70 | 2.15% | 5101 | 207.4 | 198.9 | 26.6 | .... | 334.9 | 15.23 |
| Independent Average | 8.15 A.M.<br>12.15 P.M. | 5233 | 70 | 2.15% | 5052 | 206.4 | 196.9 | 26.6 | 747 | 333.0 | 15.17 |

Number of nozzles open, seven (7).
Average reading of barometer, 29.90 in.
Average temperature of room, 97° F.

| Date, 1902. | Hour. | Feed-water Weighed per Hour, Lbs. | Condensation from Separator, Lbs. | Moisture in Steam at Throttle by Calorimeter. | Dry Steam Entering Turbine, Lbs. | Pressure above Governor-valve, Lbs. | Pressure below Governor-valve, Lbs. | Vacuum, In. | R.P.M. of Generators. | Brake Horse-power. | Dry Steam used per Brake H.P. per Hour, Lbs. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| May 23<br>May 23 | 12.45 P.M.<br>1.45 P.M. | 4675 | 60 | 2.15% | 4516 | 207.0 | 196.6 | 26.8 | ... | 284.4 | 15.88 |
| May 23<br>May 23 | 1.45 P.M.<br>2.45 P.M. | 4499 | 60 | 2.15% | 4344 | 207.7 | 196.4 | 26.8 | ... | 285.2 | 15.23 |
| Independent Average | 12.45 P.M.<br>2.45 P.M. | 4587 | 60 | 2.15% | 4430 | 207.3 | 196.5 | 26.8 | 746 | 284.8 | 15.56 |

Number of nozzles open, five (5).
Average reading of barometer, 29.83 in.
Average temperature of room, 97° F.

| Date, 1902. | Hour. | Feed-water Weighed per Hour, Lbs. | Condensation from Separator, Lbs. | Moisture in Steam at Throttle by Calorimeter. | Dry Steam Entering Turbine, Lbs. | Pressure above Governor-valve, Lbs. | Pressure below Governor-valve, Lbs. | Vacuum, In. | R.P.M. of Generators. | Brake Horse-power. | Dry Steam used per Brake H.P. per Hour, Lbs. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| May 23<br>May 23 | 3.00 P.M.<br>4.00 P.M. | 3483 | 51 | 2.15% | 3358 | 207.5 | 196.5 | 27.3 | .... | 194.8 | 17.24 |
| May 23<br>May 23 | 4.00 P.M.<br>5.00 P.M. | 3219 | 51 | 2.15% | 3100 | 207.8 | 195.1 | 27.4 | .... | 195.6 | 15.85 |
| Independent Average | 3.00 P.M.<br>5.00 P.M. | 3351 | 51 | 2.15% | 3229 | 207.6 | 195.8 | 27.35 | 751 | 195.2 | 16.54 |

Number of nozzles open, three (3).
Average reading of barometer, 29.81 in.
Average temperature of room, 80° F.

| Date, 1902. | Hour. | Feed-water Weighed per Hour, Lbs. | Condensation from Separator, Lbs. | Moisture in Steam at Throttle by Calorimeter. | Dry Steam Entering Turbine, Lbs. | Pressure above Governor-valve, Lbs. | Pressure below Governor-valve, Lbs. | Vacuum, In. | R.P.M. of Generators. | Brake Horse-power. | Dry Steam used per Brake H.P. per Hour, Lbs. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| June 10<br>June 10 | 6.35 P.M.<br>7.35 P.M. | 1996 | 33 | 2.15% | 1921 | 201.1 | 196.5 | 28.1 | .... | 115.0 | 16.70 |
| June 10<br>June 10 | 7.35 P.M.<br>8.35 P.M. | 2098 | 33 | 2.15% | 2021 | 201.6 | 198.9 | 28.1 | .... | 122.0 | 16.57 |
| June 10<br>June 10 | 8.35 P.M.<br>9.35 P.M. | 1984 | 33 | 2.15% | 1909 | 201.7 | 198.4 | 28.1 | .... | 121.5 | 15.71 |
| Independent Average | 6.35 P.M.<br>9.35 P.M. | 2026 | 33 | 2.15% | 1950 | 201.5 | 197.9 | 28.1 | 751 | 118.9 | 16.40 |

All barometer readings are reduced to 32° F.

## Tests with Superheated Steam.

Number of nozzles open, eight (8).
Average reading of barometer, 30.18 in.
Average temperature of room, 83° F.

| Date, 1902. | Hour. | Weight of Steam Used per Hour, Lbs. | Pressure above Governor-valve, Lbs. | Pressure below Governor-valve, Lbs. | Vacuum, Ins. | Superheat Governor-valve. | Revs. per Minute of Generators. | Brake Horse-power. | Steam Used per Brake Horse-power per Hour, Lbs. |
|---|---|---|---|---|---|---|---|---|---|
| May 22 | 8– 9 A.M. | 4833 | 208.3 | 200.6 | 27.2 | 81° F. | ...... | 356.6 | 13.55 |
| May 22 | 9–10 A.M. | 4936 | 207.5 | 199.3 | 27.2 | 86° F. | ...... | 355.7 | 13.88 |
| May 22 | 10–11 A.M. | 5083 | 207.7 | 202.1 | 27.2 | 91° F. | ...... | 357.8 | 14.21 |
| May 22 | 11–12 A.M. | 4976 | 208.3 | 199.4 | 27.2 | 88° F. | ...... | 354.1 | 14.05 |
| May 22 | 12– 1 P.M. | 4841 | 207.5 | 194.3 | 27.3 | 82° F. | ...... | 343.5 | 14.09 |
| May 22 | 1– 2 P.M. | 4768 | 206.9 | 195.6 | 27.2 | 75° F. | ...... | 344.4 | 13.84 |
| Independent Average | 8– 2 P.M. | 4906 | 207.0 | 198.5 | 27.2 | 84° F. | 750 | 352.0 | 13.94 |

Number of nozzles open, seven (7).
Average reading of barometer, 30.07 in.
Average temperature of room, 90° F.

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| May 22 | 2.10 P.M. | 4316 | 207.5 | 196.2 | 27.4 | 67° F. | ...... | 299.8 | 14.39 |
| May 22 | 3.10 P.M. | | | | | | | | |
| May 22 | 3.10 P.M. | 4248 | 207.3 | 197.9 | 27.4 | 61° F. | ...... | 297.3 | 14.29 |
| May 22 | 4.10 P.M. | | | | | | | | |
| Independent Average | 2.10 P.M. – 4.10 P.M. | 4282 | 207.4 | 197.0 | 27.4 | 64° F. | 756 | 298.4 | 14.35 |

Number of nozzles open, five (5).
Average reading of barometer, 29.79 in.
Average temperature of room, 89° F.

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| June 10 | 8.45 A.M. | 3068 | 199.2 | 196.5 | 27.6 | 8° F. | ...... | 195.3 | 15.71 |
| June 10 | 9.45 A.M. | | | | | | | | |
| June 10 | 9.45 A.M. | 3010 | 201.5 | 197.2 | 27.4 | 12° F. | ...... | 197.3 | 15.26 |
| June 10 | 10.45 A.M. | | | | | | | | |
| June 10 | 10.45 A.M. | 3020 | 201.4 | 196.1 | 27.4 | 10° F. | ...... | 196.5 | 15.37 |
| June 10 | 11.45 A.M. | | | | | | | | |
| Independent Average | 8.45 A.M. – 11.45 A.M. | 3033 | 200.7 | 196.6 | 27.5 | 10° F. | 743 | 196.5 | 15.44 |
| June 10 | 1.45 P.M. | 3107 | 201.4 | 196.7 | 27.4 | 13° F. | ...... | 194.8 | 15.95 |
| June 10 | 2.45 P.M. | | | | | | | | |
| June 11 | 2.45 P.M. | 3054 | 203.1 | 199.0 | 27.3 | 15° F. | ...... | 197.9 | 15.43 |
| June 10 | 3.45 P.M. | | | | | | | | |
| June 10 | 3.45 P.M. | 3025 | 202.7 | 197.5 | 27.4 | 19° F. | ...... | 194.7 | 15.54 |
| June 10 | 4.45 P.M. | | | | | | | | |
| Independent Average | 1.45 P.M. – 4.45 P.M. | 3062 | 202.4 | 197.7 | 27.4 | 16° F. | 747 | 196.0 | 15.62 |
| Average of both tests.............................. | | | | | | | 745 | ..... | 15.53 |

The following table* gives results of acceptance tests made upon a 300-horse-power De Laval turbine in November, 1904:

*Machine No. 2083.* *Nov. 18, 1904.*

RESULT SHEET.

| | | | | | |
|---|---|---|---|---|---|
| Run No. | 1 | 2 | 3 | 4 | 5 |
| Duration of run, minutes | 55 | 55 | 55 | 55 | 55 |
| Revolutions of generator-shafts per minute | 907 | 897 | 900 | 898 | 895 |
| Steam-pressure above governor-valve, pounds, gage | 152 | 152 | 152 | 152 | 152 |
| Steam-pressure below governor-valve | 133 | 144 | 140 | 136 | 140 |
| Load in per cent of rated load | $\frac{1}{4}$ | $\frac{1}{2}$ | $\frac{3}{4}$ | 1 | $1\frac{1}{4}$ |
| Vacuum, inches mercury | 27.25 | 27.19 | 26.85 | 26.20 | 25.75 |
| Back pressure, pds. sq. in. abs. | 1.60 | 1.64 | 1.84 | 2.14 | 2.36 |
| Number of nozzles open | 4 | 5 | 7 | 9 | 10 |
| Quality of steam, per cent | 100 | 100 | 100 | 100 | |
| Superheat of steam, deg. F. | ....... | ....... | ....... | ....... | 19.9 |
| Total D. H. P. | 98.1 | 159.5 | 236 | 302.5 | 348 |
| " steam per hour, pounds | 2286.4 | 3049.0 | 4183.9 | 5326.5 | 6145.0 |
| Steam per D. H. P. hour, pounds | 23.3 | 19.1 | 17.71 | 17.6 | 17.64 |
| Total K.W. | 56.63 | 100.38 | 155.2 | 201.13 | 233.2 |
| Steam per K.W. hour, pounds | 40.4 | 30.35 | 26.95 | 26.5 | 26.35 |

**The Parsons Turbine** embodies a combination of the impulse and reaction principles. The steam expands during its passage through the Parsons turbine much as it does in an expanding nozzle; that is, the cross-sectional area of steam-passage increases from the high- to the low-pressure end of the turbine, according to the volume and velocity of the steam at the various points of its path. The annular space between the stationary casing and the rotating spindle corresponds essentially to a simple steam-nozzle, with the difference that in a nozzle the heat energy is expended upon the steam itself in producing high velocities of efflux; whereas, in the turbine, the kinetic energy of the steam due to the heat drop in any one stage is expended in producing rotation of the spindle.

The heat given up in any one stage is limited to that amount which will produce the kinetic energy desired to be absorbed in that stage. Further increments of heat drop in succeeding stages add successive increments of rotative effort to the spindle,

* Thesis test of Messrs. Crosier and Little, Sibley College, 1905.

until, when the steam has passed entirely through the turbine, it has fallen in temperature and pressure an amount corresponding to the total heat given up as work, plus the losses experienced in the machine.

The fact that the heat drop is divided into a great number of steps, the energy being absorbed as rotative effect during each step, causes the steam velocity to be kept low throughout the machine, and allows a comparatively low peripheral speed of blades to be employed with good efficiency.

The general arrangement and various details of the Parsons turbine, as manufactured by the Westinghouse Machine Company, are shown in Figs. 93–100.

The curves in Fig. 92 show economy attained by the use of saturated steam and superheated steam, and the effect of increase of vacuum.

The table below gives the trial results represented by the

| Gage Pressure. | | Degrees Super-heat, F. | Vacuum, Inches of Mercury. | Brake Horse-power. | Steam Consumption. | | | |
|---|---|---|---|---|---|---|---|---|
| Before Entering Throttle. | After passing Throttle. | | | | Per Hour, Total. | Gland Leakage. | Net per Hour. | Per B.H.P. Hour. |
| 152 | About | 97 | 27.3 | 269 | 4352 | 339 | 4013 | 14.9 |
| 150 | 130 | 85 to 105 | 27.3 | 402 | 5883 | 349 | 5534 | 13.7+ |
| 150 | to | 100 | 27.3 | 649 | 8558 | 343 | 8310 | 12.8 |
| 152 | 120 | 95 | 27.3 | 766 | 10062 | 406 | 9656 | 12.6 |
| 150 | | 100 | 26.9 | 956 | 12858 | 465 | 12393 | 12.95 |
| 150 | | 105 | 26.6 | 1195 | 16820 | 453 | 16367 | 13.7 |
| | About | | | | | | | |
| 150 | | None | 27.3 | 245 | 4765 | 295 | 4470 | 18.2 |
| 150 | | " | 27.3 | 406 | 6628 | 310 | 6261 | 15.4 |
| 150 | 117 | " | 27.3 | 650 | 9490 | 347 | 9175 | 14.1 |
| 150 | 129 | " | 27.3 | 716 | 10488 | 394 | 10122 | 14.1 |
| 150 | 138 | " | 26.3 | 1144 | 18015 | 387 | 17650 | 15.4 |

curves plotted in Fig. 92. The turbine was of 400 K.W. rated capacity, equipped with automatic by-pass valve. Revolutions per minute 3600. The results are intended to show the gain in economy due to the use of superheated steam. It is to be

noted that the vacuum was only about 27 inches of mercury. The power was absorbed by a water-brake.

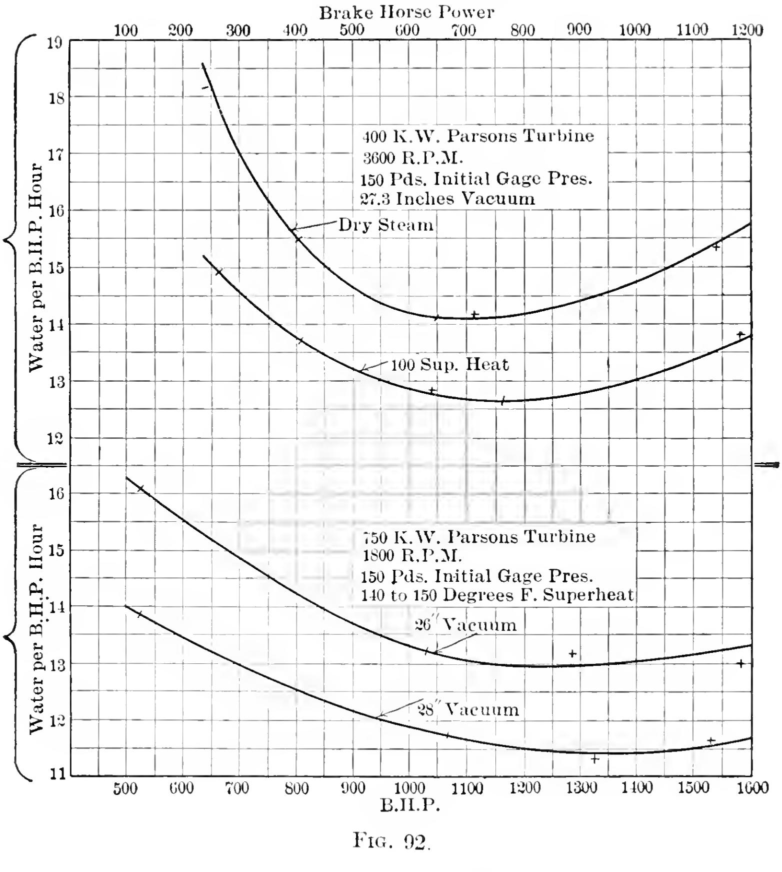



Fig. 92.

The table on page 252 gives the trial results, represented in Fig. 92, from a 750-K.W. Parsons turbine running at 1800 revolutions per minute. The power was absorbed by a water-brake. The results are intended to show the gain in economy obtained by increasing the vacuum from 26″ to 28″. All the tests were made with superheated steam.

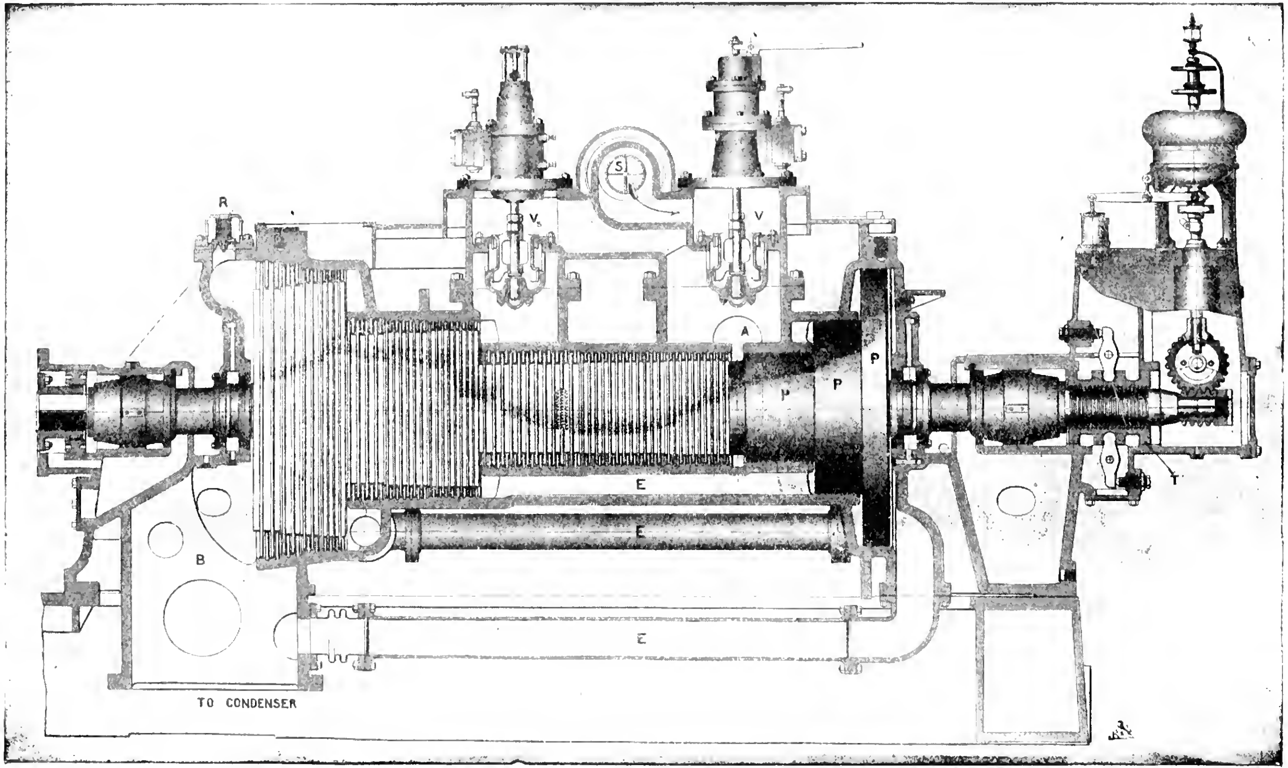



FIG. 93.—Cross-sectional view of a Westinghouse-Parsons steam-turbine.

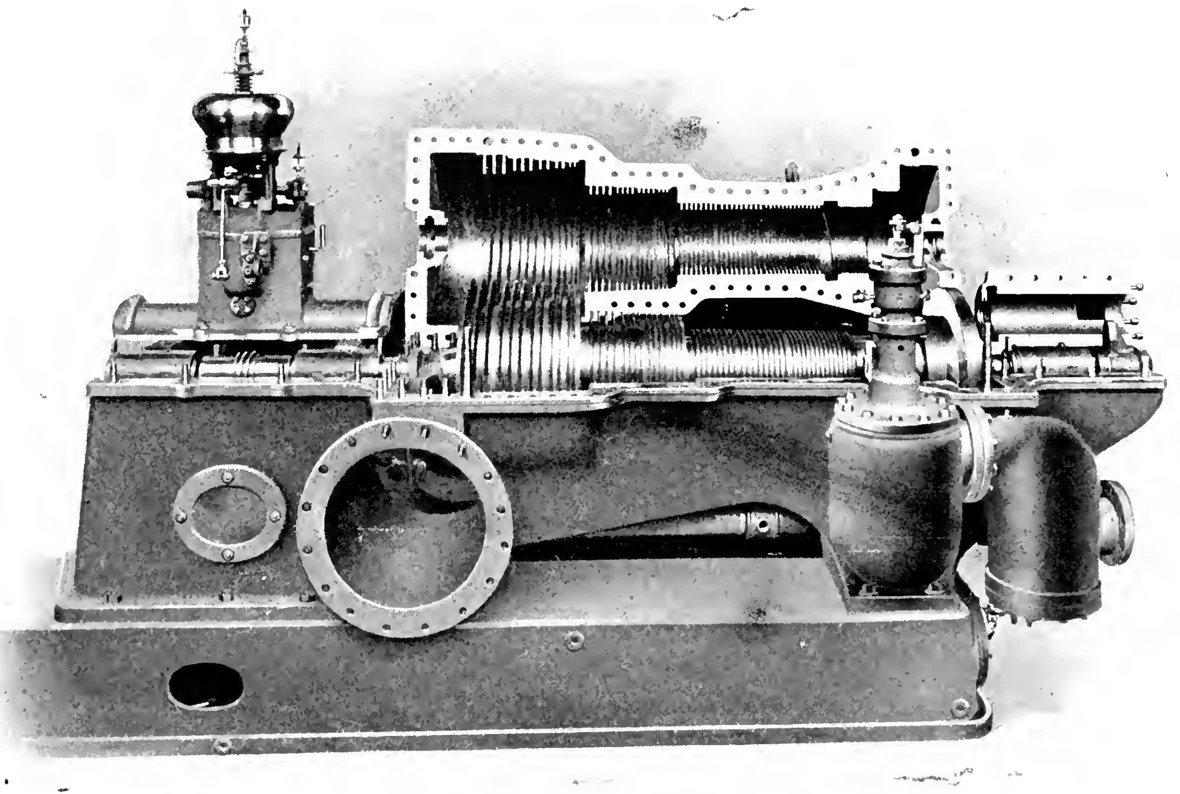


FIG. 94.—400-K.W. Westinghouse-Parsons turbine. 3600 R.P.M.

| Gage Pressure. | | Vacuum, Inches. | Degrees Superheat. | Brake Horse-power. | Steam Consumption. | |
|---|---|---|---|---|---|---|
| Before Entering Throttle. | After Passing Throttle. | | | | Total per Hour. | Per B.H.P. Hour. |
| 153 | 62 | 26 | 150 | 524 | 8459 | 16.04 |
| 152 | 108 | 26 | 146 | 1025 | 13516 | 13.18 |
| 150 | 136 | 26 | 147 | 1285 | 16784 | 13.14 |
| 146 | 122 | 26 | 142 | 1586 | 19938 | 12.98 |
| 152 | 51 | 28 | 144 | 520 | 7194 | 13.85 |
| 149 | 102 | 28 | 153 | 1067 | 12578 | 11.79 |
| 151 | 125 | 28 | 152 | 1346 | 15368 | 11.42 |
| 150 | 138 | 28 | 153 | 1530 | 17623 | 11.52 |

**The essential difference between the impulse- and the reaction-turbines** is, that in the former the pressures on the two sides of a rotating wheel and also of a guide-wheel are

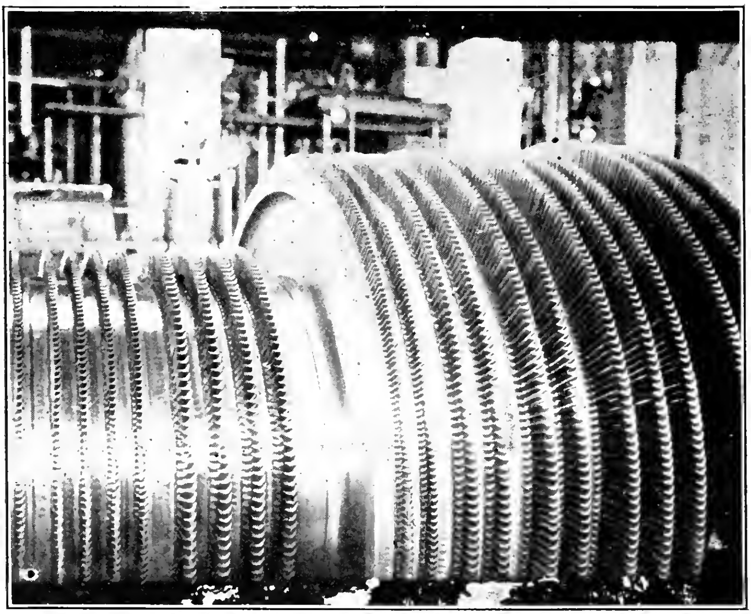


FIG. 95.—Blading of a Westinghouse-Parsons steam-turbine. Rotor only.

equal to each other, or are supposed to be in the ideal case; in the latter the pressure drops from the entering side of either a guide or a rotating wheel, and thus expansion and acceler-

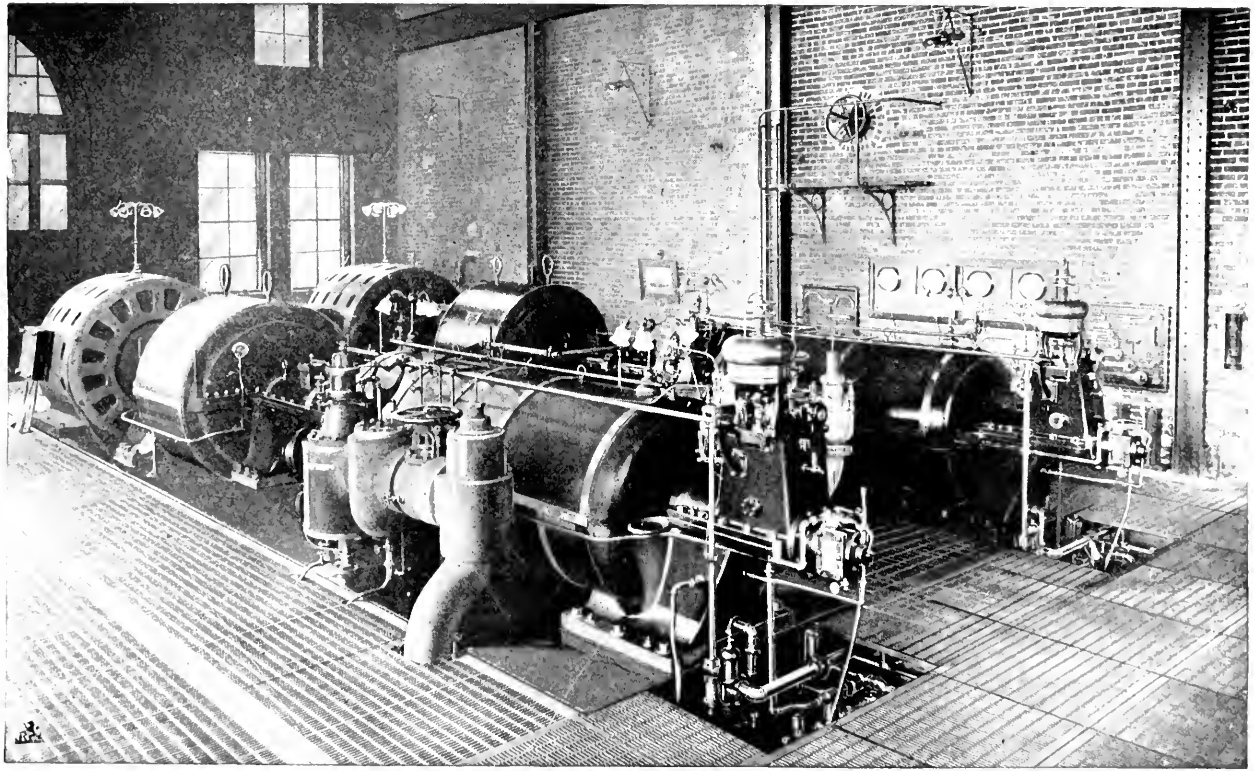


FIG 96.—Westinghouse-Parsons turbines and generators.

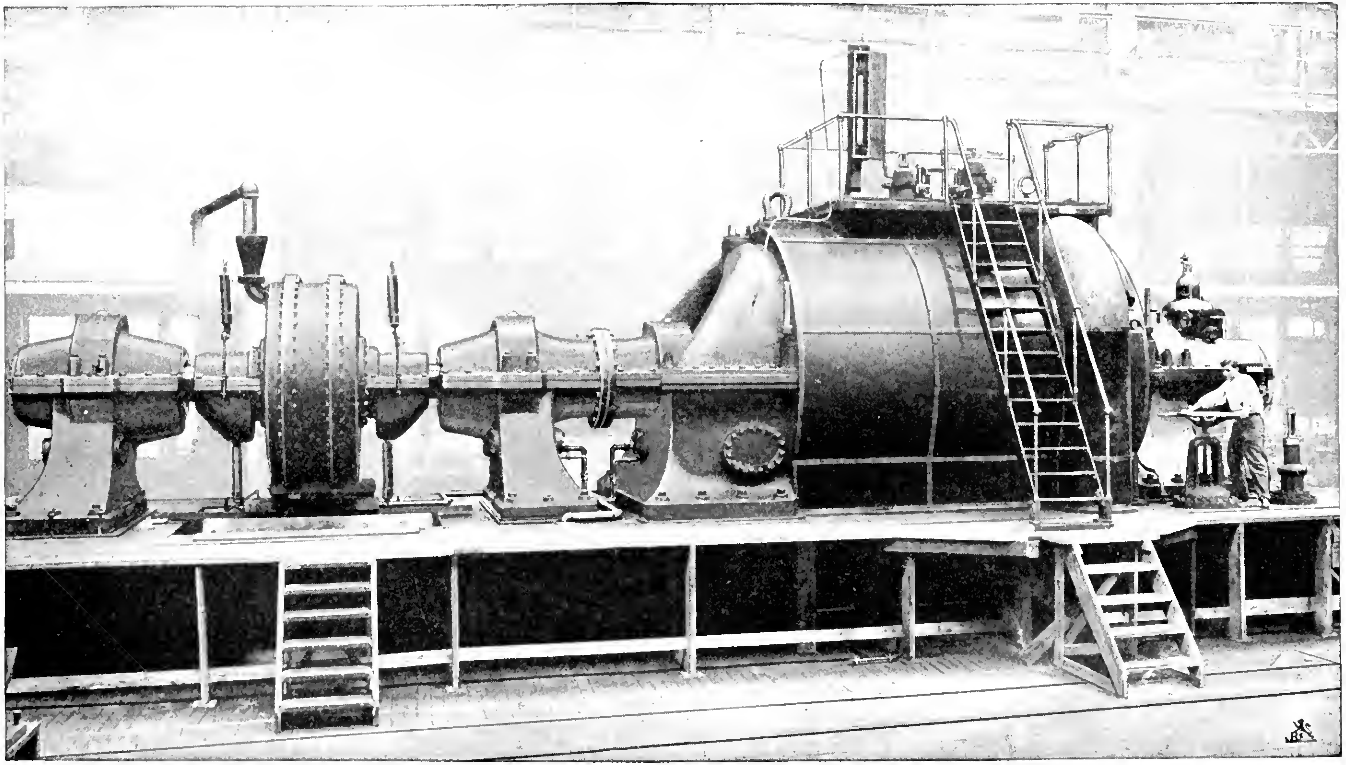


FIG. 97.—Westinghouse-Parsons 5500-K.W. turbine, connected to water-brake for testing at shop.

ation of the jet occur in each row of blades. In the simple impulse-turbine a set of nozzles discharges upon the buckets

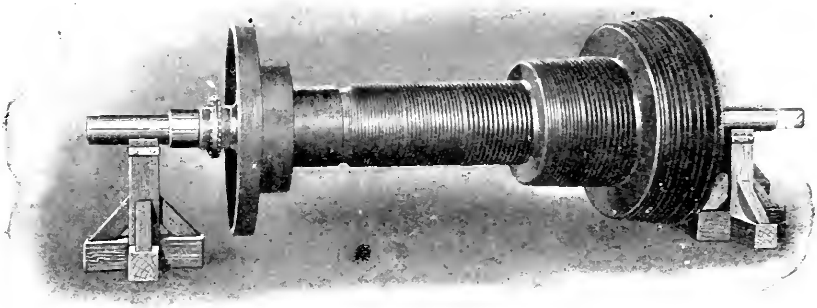


FIG. 98.—Rotor, complete, with balance-pistons, Westinghouse-Parsons turbine.

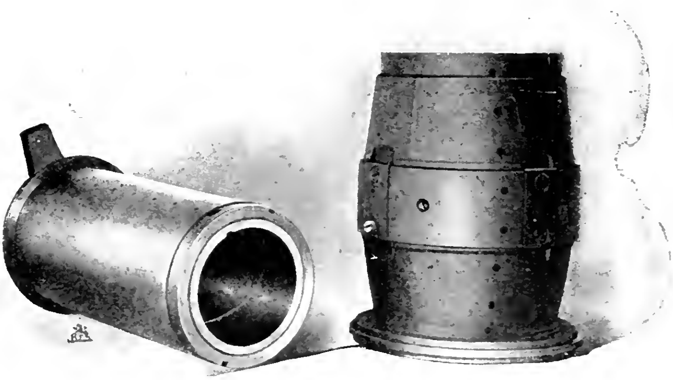


FIG. 99.—Bearing, with concentric brass sleeves, Westinghouse-Parsons turbine.

of a single wheel. In the compound impulse-turbine a set of nozzles discharges upon a series of moving and stationary rows of buckets, the latter changing the direction from the

former, so that rotation in a common direction is produced by the action of the steam upon each moving wheel. The discharge from any set of rotating and guide wheels may be allowed to expand through a second set of nozzles, or orifices, and the resulting jet caused to act upon a second series of rotating and guide wheels. The same process may be repeated in succeeding stages, to as great an extent as necessary to absorb as much as possible of the energy of the steam.

In the Parsons turbine as applied to stationary work, the end thrust caused by the axial component of the action of the steam on the blades is neutralized so as to prevent the spindle moving in an axial direction, by balance-pistons, as shown at *P* in Fig. 93. These are grooved at the periphery, and mesh with corresponding grooves and projections on the stationary part of the machine so as to prevent leakage of the steam past them. The area of the pistons is proportioned according to the amount of thrust which they are required to balance.

For the low-pressure cylinders of Parsons turbines the blades become quite long, and in order to give them sufficient stiffness special means are taken for holding the outer ends of the blades. The Westinghouse Machine Company employs for this purpose a special form of wire "lacing," which holds the ends of the blades firmly. The recess in the largest blade shown in Fig. 102 is for receiving the stiffening-strip or shroud-wire.

In the turbines for the large Cunard steamer "Carmania" this form of fastening was tried first, but was modified because the expansion of the turbine parts required that the ends of the blades should be held less rigidly. The modification consisted in making the shroud-wire in sections, and joining the ends by inserting them in short lengths of tubing, flattened so they took the place of the shroud-wire at certain places in the circumference. The shroud-wire was thus provided with slip-joints, as the ends were free to move back and forth in the flattened tubes.

In Parsons turbines of small sizes flexible bearings are used in order to permit the spindle to revolve about its gravity instead of its geometric axis, so that at high speeds the effect of minute errors in balancing of the disks may be neutralized. The flexible bearings consist of several concentric bronze sleeves, with sufficient clearance to allow oil-films to form between the sleeves, thus permitting the shaft to vibrate within narrow limits. In all machines running below 1200 revolu-

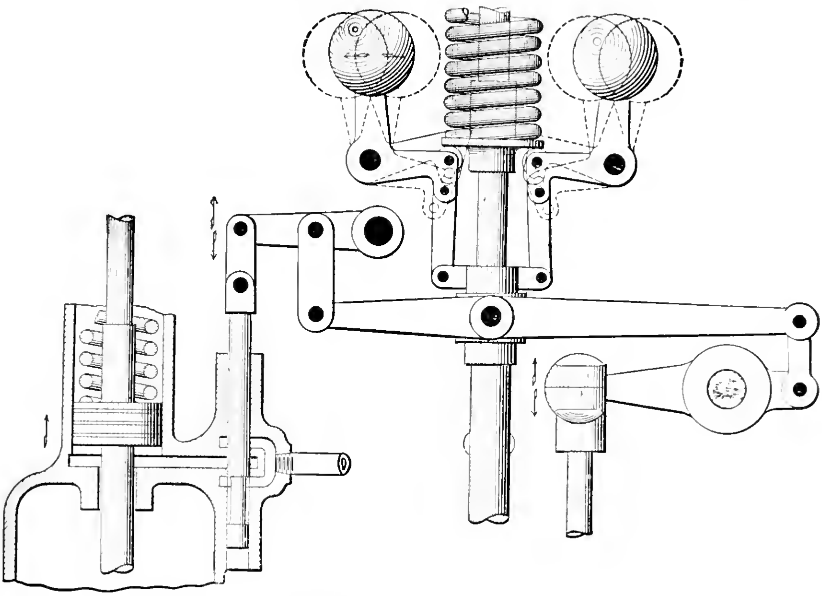


FIG. 100.—Westinghouse-Parsons governor and connections to controlling-valve.

tions per minute, however, the flexible bearing is replaced by a solid self-aligning bearing.

Water-sealed packing-glands are used at the ends of the casings to prevent the escape of steam or the influx of air at the point of entry of the shaft.

Steam enters the turbine through a strainer, thence through a poppet-valve controlled by the governor. In the manner of operating this valve, practice varies among the different makers of Parsons turbines. As made by the Westinghouse Machine Company, the poppet-valve opens and closes at inter-

vals proportional to the speed of the turbine. At light loads the valve opens for short periods, remaining closed the greater part of the time. As the load increases the valve remains open longer, until when full pressure is continually maintained in the high-pressure end of the turbine the valve merely vibrates without sensibly affecting the pressure of the steam. If the load on the machine is still further increased, an auxiliary poppet-valve begins to open, and admits steam from the throttle-valve directly into the lower cylinders of the turbine, increasing the total power developed. The economy decreases with the opening of this secondary or "by-pass" valve, but the range of load at which the turbine may be operated with a fair degree of economy is very greatly extended. The intermittent action of the valve admitting the steam is accompanied by a constantly reciprocating motion of the operating mechanism, which is thereby made especially sensitive.

The bearings of Parsons turbines are supplied with oil under pressure, a continuous stream being circulated by an oil-pump operated from the main shaft.

**The Allis-Chalmers Company** of Milwaukee has recently entered the steam-turbine field with a turbine of the Parsons type, with the arrangement of blading shown in Figs. 105 to 107. The roots of the blades are formed in dovetail shape, and inserted in slots, cut in foundation- or base-rings, the slots conforming to the shape of the blade-roots. The foundation-rings are of dovetail cross-section, and are inserted in dovetailed grooves, cut in the turbine spindle and cylinder respectively, in which they are firmly held by key-pieces. The latter, after being driven into place, are upset into undercut grooves. The tips of the blades are protected and reinforced by a shouldered projection, which is inserted in a slot, punched in a shroud-ring. These slots are so punched as to produce accurate spacing, and at the same time to give the proper angles to the blades, independent of the slots in the base-ring. After insertion in the slots, the blade-tips are riveted over.

The shroud-rings are made in channel shape, with thin projecting flanges. This is to protect the ends of the blades in case of accidental contact, and at the same time is thought to reduce the loss by leakage. The blading is put in place and fastened by machinery.

There is, in this type of Parsons turbine, a special arrangement of balance-piston, placed in the low-pressure end instead of in the high-pressure end of the turbine, and leakage past it is prevented by what is called a *labyrinth-packing*, consisting of radial baffles. The construction and general arrangement of the turbine is shown in Figs. 103–107.

**The meaning of the word "stage"** in the two types of turbine has been variously defined. In the impulse-turbine a *stage* consists of a set of nozzles and a set of buckets upon which the jet from the nozzles acts. If, as in the case of the Curtis turbine, and others of the same type, the discharge from the first moving buckets is guided into succeeding moving buckets, in order to absorb further the kinetic energy which has been produced in the nozzles, the whole combination of nozzles and the wheels upon which the jet acts is called a *stage*. If a second set of nozzles be added, discharging upon one or more moving wheels, this becomes the *second stage* of the turbine, and so on.

In the Parsons turbine, since the stationary or guide blades, in one row, act as nozzles for the succeeding row of moving blades, the two rows taken together may be correctly called a *stage*. Exception has been taken to this, upon the ground that expansion occurs in the moving as well as in the guide blades, and it has therefore been suggested that each row of moving blades and each row of guide-blades form a complete stage. Throughout this book the word *stage*, as applied to the Parsons type of turbine, means a row of guide-blades and a row of moving blades taken together.

The elements upon which the steam acts in impulse-turbines are commonly called *buckets*, a name used in connection with water-wheels. In turbines of the Parsons type the elements

acted upon by the steam are of quite different shape, and of g. eater length than those in the impulse-turbine, and are known as *blades*, or sometimes as *vanes*. Fig. 102 shows various sizes of blades, as used in Parsons turbines; and on page 163 are shown outlines of the buckets used in the Curtis turbine.

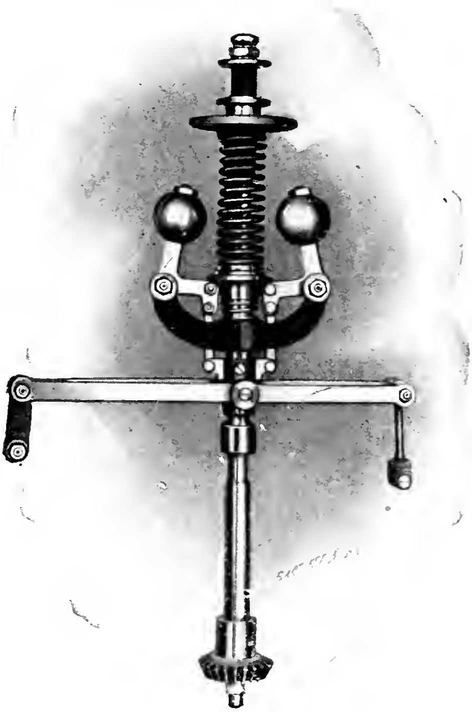


FIG. 101.—Westinghouse-Parsons governor.

**The Compound Impulse-turbine.** — The best known turbine of the compound impulse type manufactured in this country is the Curtis. Figs. 109–118 show general arrangements and structural details of the machine as manufactured by the General Electric Company.

As shown in Fig. 60 illustrating the 500-K.W. two-stage machine, the turbine proper is divided into two compartments, in each of which are three moving bucket-wheels and two rows of

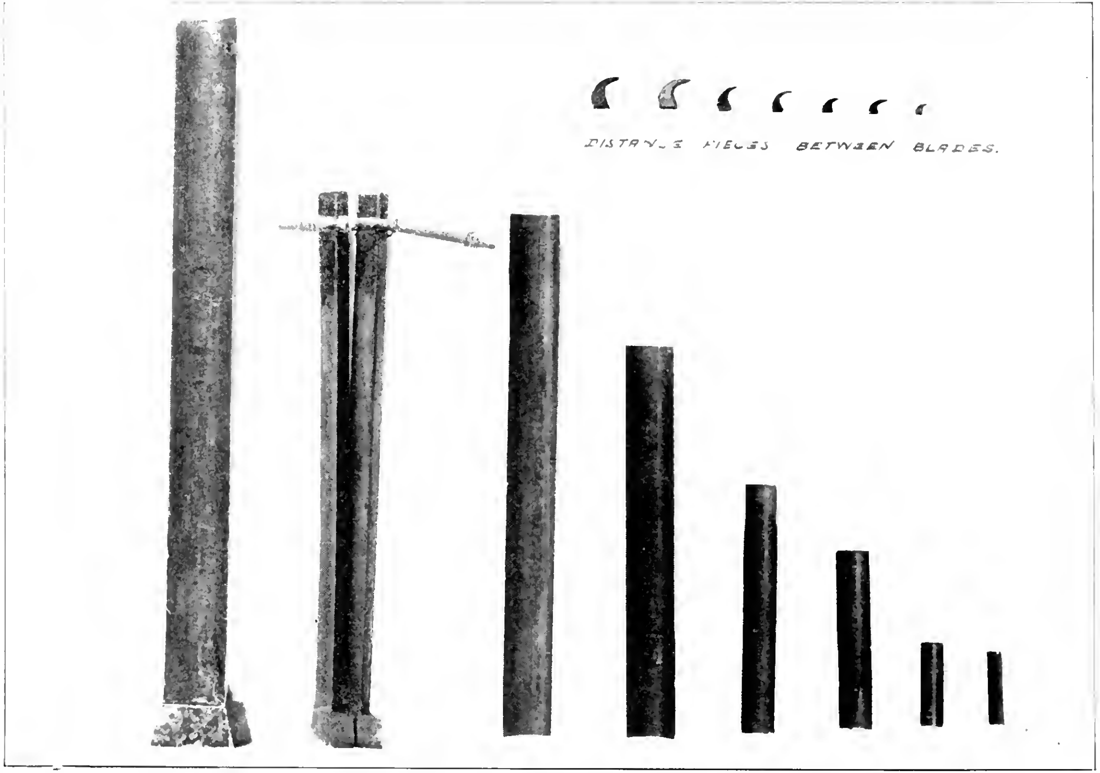



FIG. 102.—Parsons turbine-blades, with calking-pieces.

stationary buckets. The three moving wheels in each stage are firmly bolted together, and are attached to a single hub mounted upon the vertical main shaft of the turbine. Before entering the buckets of the first stage the steam passes through

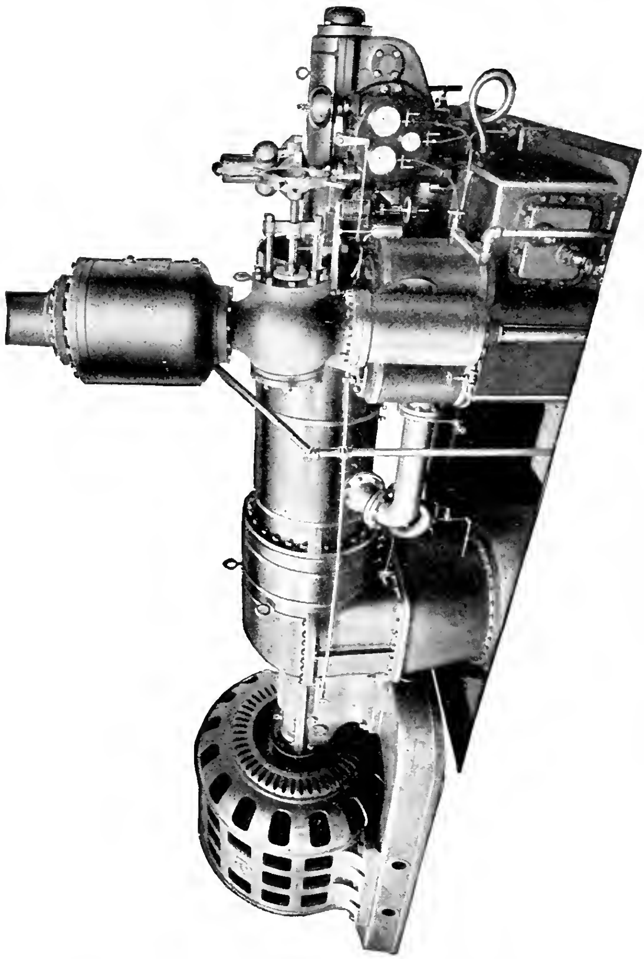


FIG. 103.—2000-K.W. Allis-Chalmers turbine and generator.

a set of twelve nozzles, about $\frac{1}{2}$ inch diameter, covering a section of the circumference about 28 inches in length. The clearance between the edges of the revolving and stationary buckets is about $\frac{1}{10}$ of an inch, and they are arranged so that there is no possibility of bucket interference.

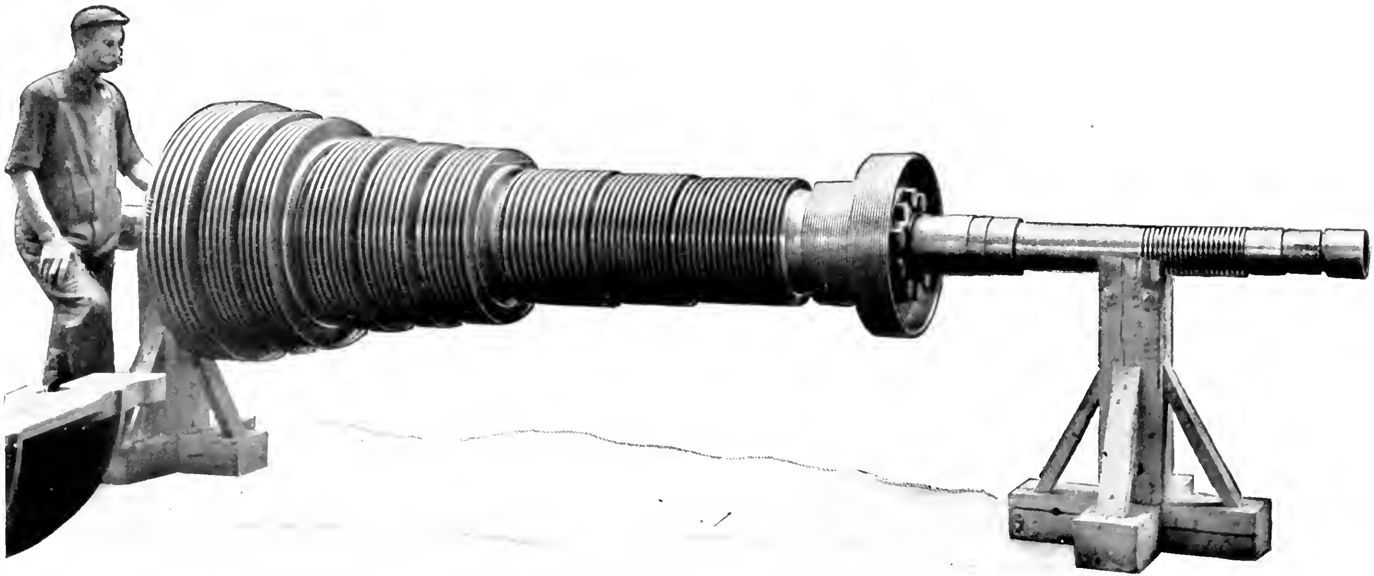


Fig. 104.—Spindle, complete, with balance-piston and thrust-collars, 1500-K.W. Allis-Chalmers turbine

The nozzles directing the steam upon the buckets of the second-stage wheels are placed in a diaphragm which separates one stage from the other. The twelve nozzles of the first

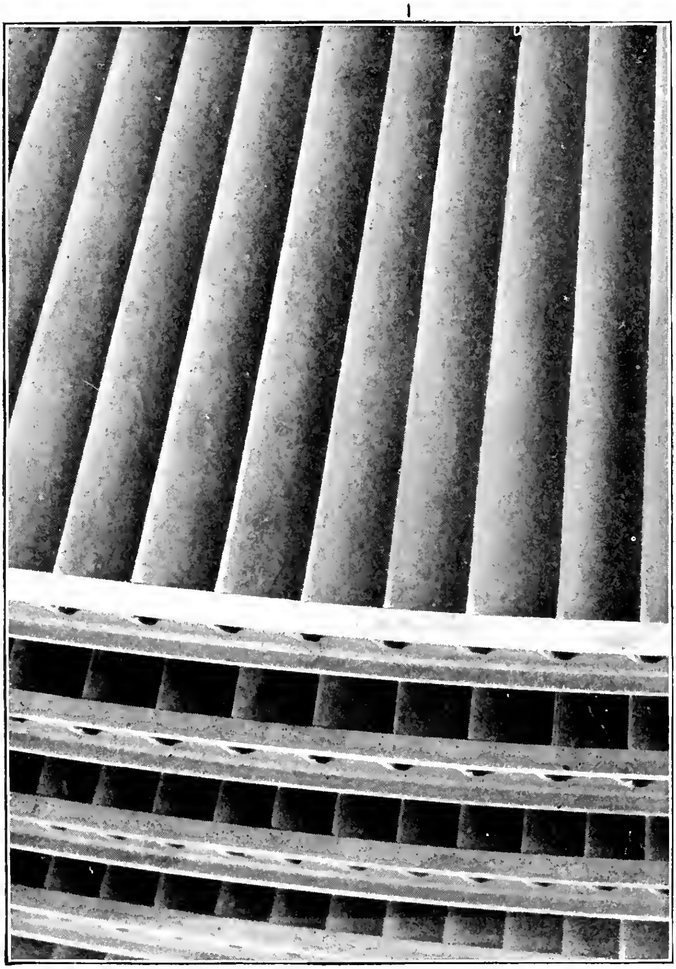


Fig. 105.—Ten-inch blades, at exhaust end of spindle, 5500-K.W. Allis-Chalmers turbine.

stage are divided into six sets, each containing two nozzles, and each set is supplied with steam through a single vertical poppet-valve. The upper end of the valve is of cylindrical shape, of larger diameter than the valve itself, and moves

FIG. 156.—Half-rings, $5\frac{1}{2}$-inch blades at exhaust end and $1\frac{5}{16}$-inch blades at inlet, for 2000-K.W. Allis-Chalmers turbine. 1800 R.P.M.

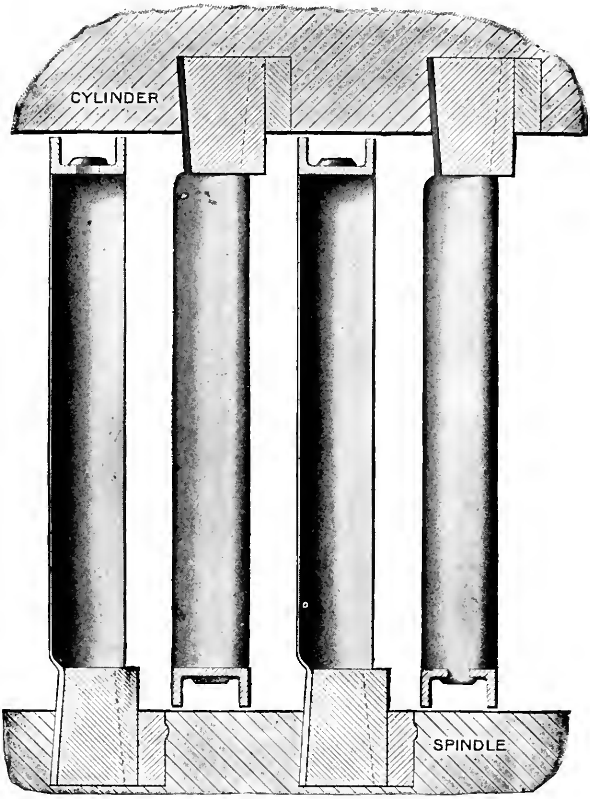



Fig. 107.—Allis-Chalmers Turbine-blading.

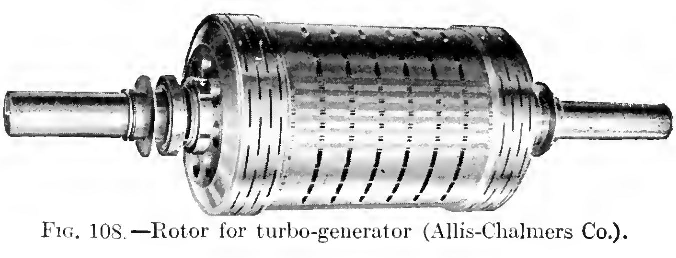


Fig. 108.—Rotor for turbo-generator (Allis-Chalmers Co.).

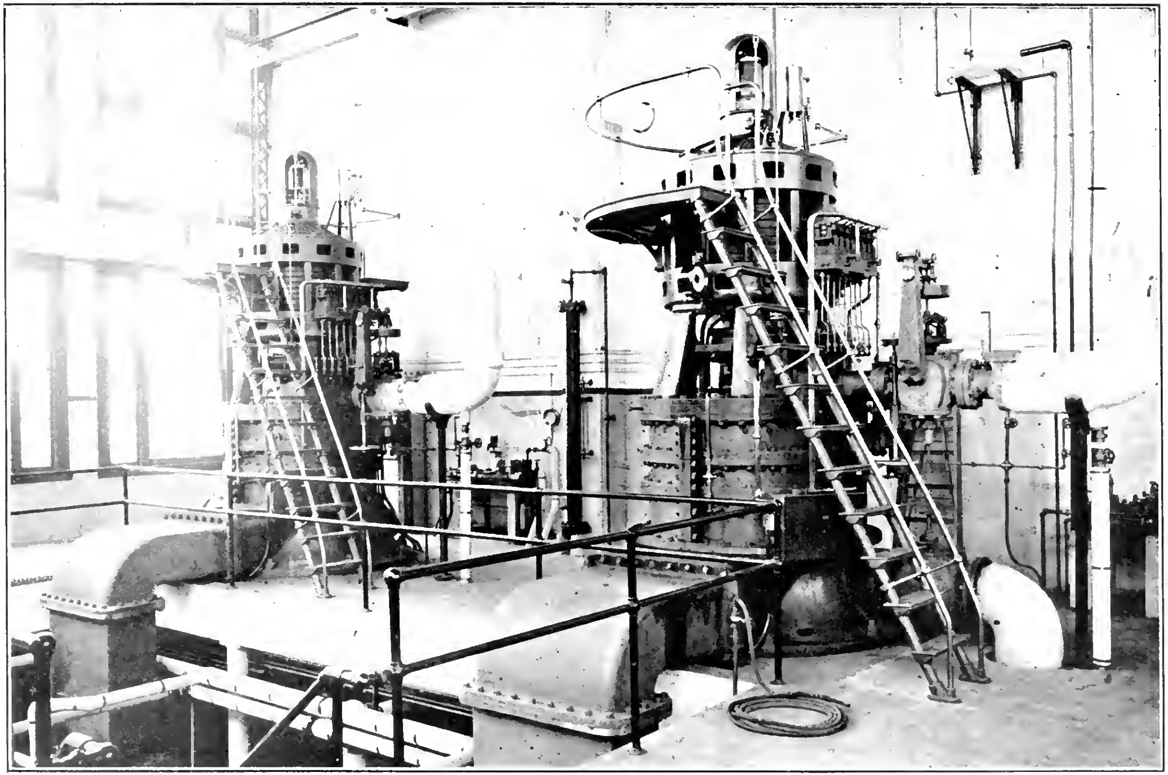


FIG. 109.—500-K.W. Curtis turbines, 1800 R.P.M., 600-volt generators.

up and down in a vertical cylinder. The valve is caused to open by steam, which is admitted through a port, opened and closed by a pilot, or "needle," valve. This pilot-valve

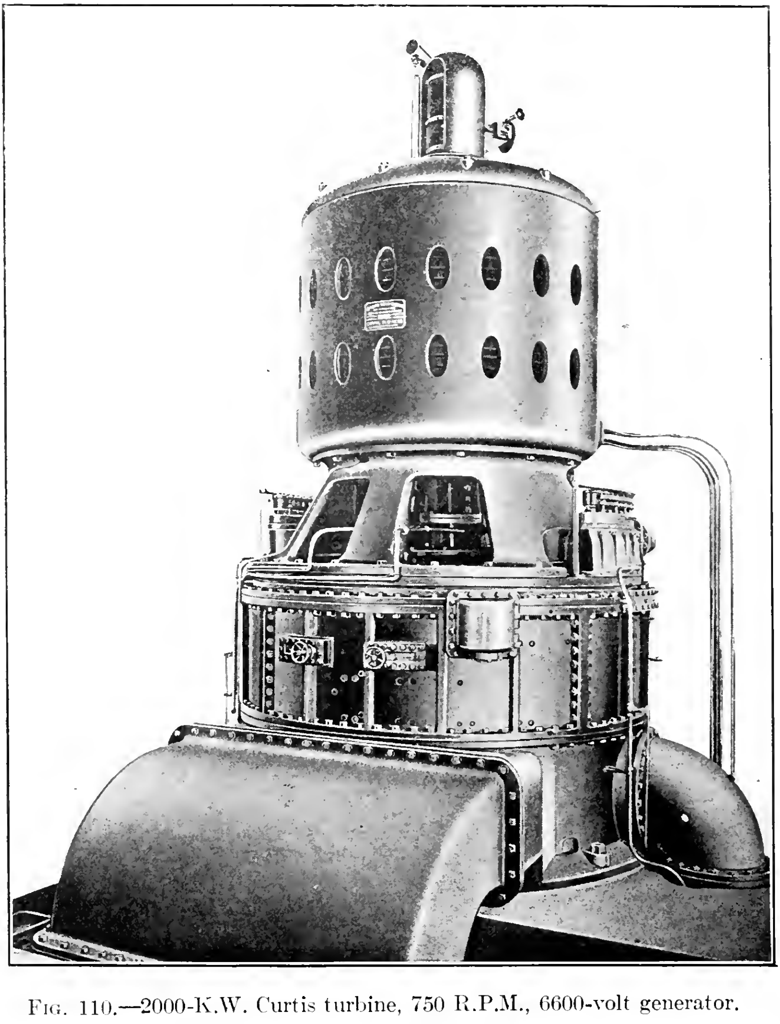


Fig. 110.—2000-K.W. Curtis turbine, 750 R.P.M., 6600-volt generator.

is actuated by an electromagnet, the circuit in which is made and broken by a controlling mechanism, which in turn is actuated by the governor at the extreme upper end of the shaft.

The number of valves which are open, and the length of time they are open, control the steam-supply, and therefore the

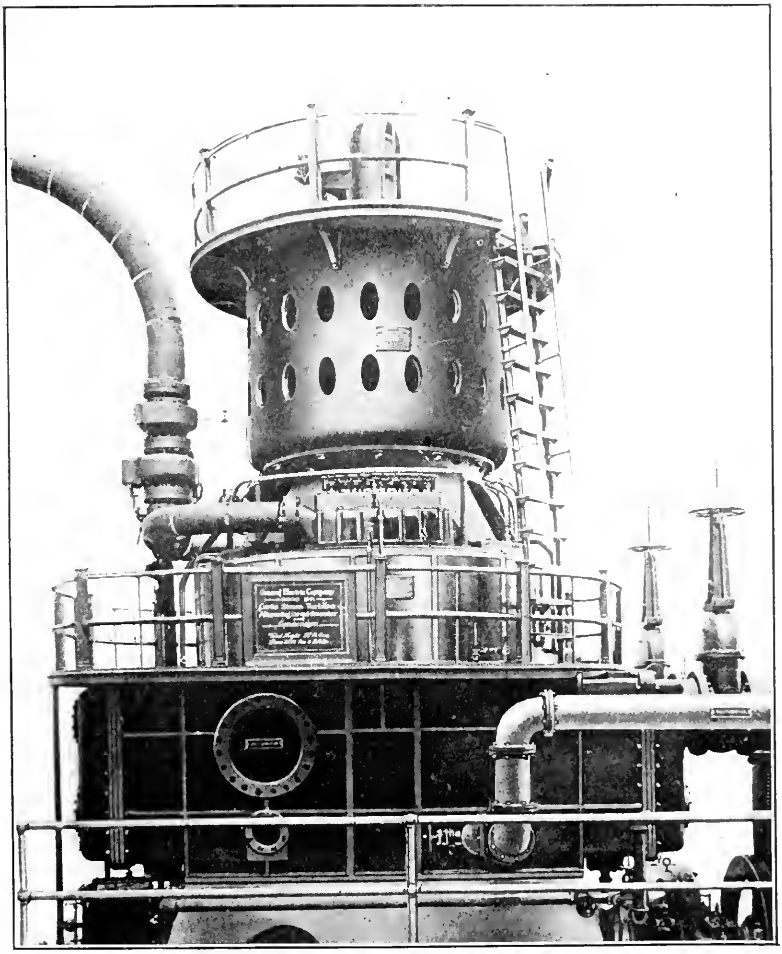


FIG. 111.—2000-K.W. Curtis turbine, four-stage, 750 R.P.M., 6600-volt generator.

power of the turbine. The valves which are operative at any one time are always either full open or completely closed, there being no intermediate position.

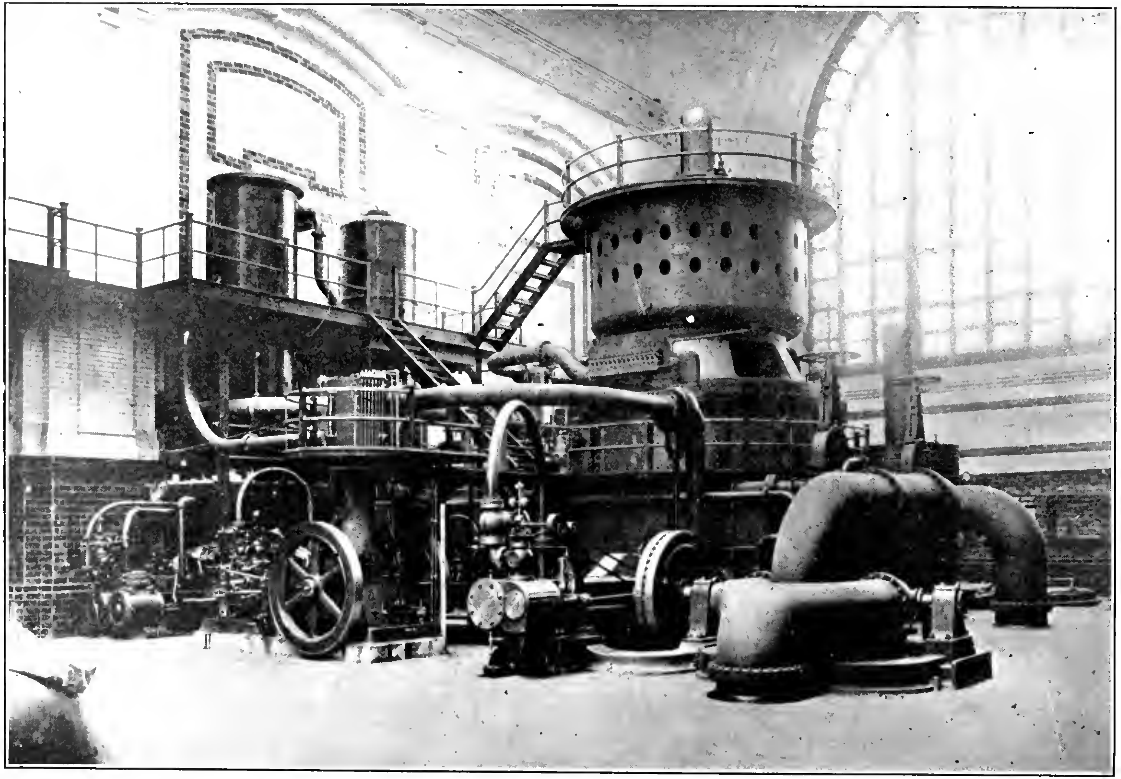


Fig. 112.—5000-K.W., four-stage Curtis turbine. 514 R.P.M., 6600-volt generator.

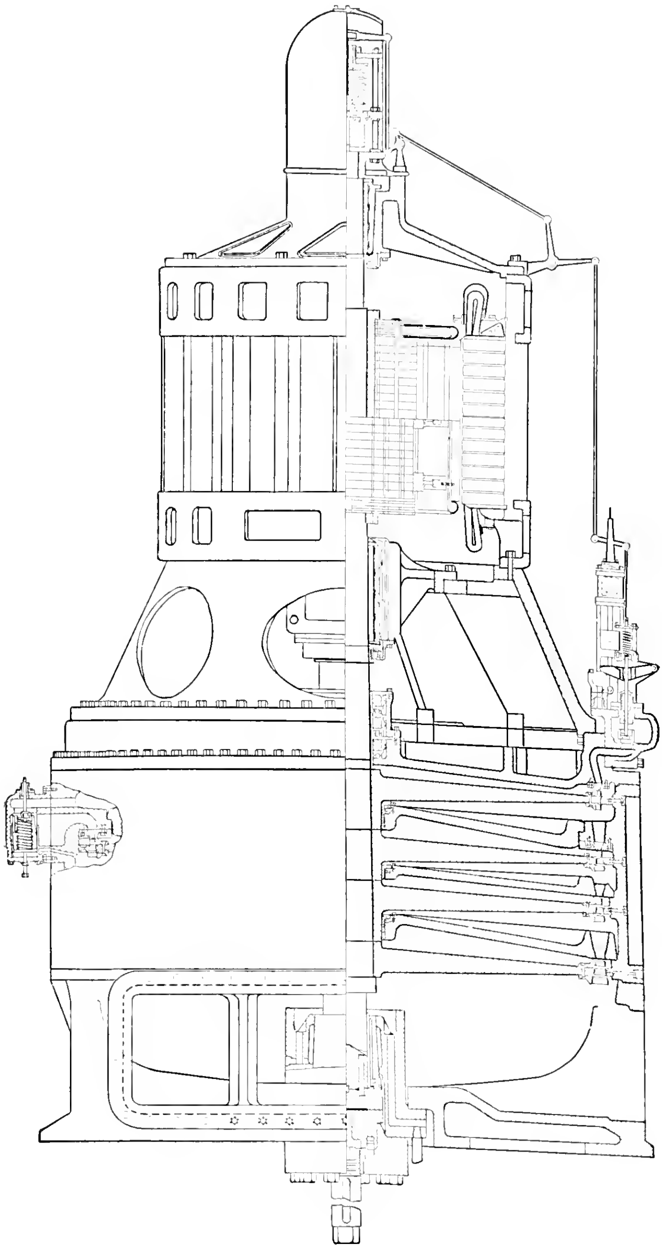


FIG. 113.—New 2000-K.W. 60-cycle turbine and generator.

Surrounding the shaft, above the first stage, and at the lower part of the second stage, are packing-boxes, which prevent leakage of air into the two chambers containing the

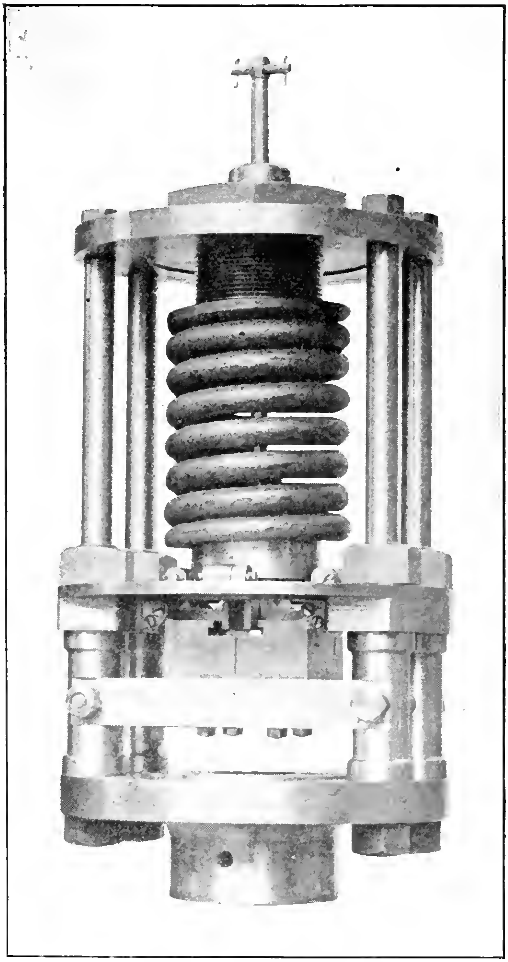


FIG. 114.—Tension-spring governor for 500-K.W. Curtis turbine.

revolving wheels. There are two carbon rings in each of these packing-boxes, which fit the shaft and the top and bottom of the packing-box closely. The space between the rings

is filled with steam, at a pressure slightly above that of the atmosphere. If any leakage should occur past the lower ring of the first-stage packing, or past the upper ring of the second-stage packing, steam would flow in and prevent the entrance of air into the turbine.

FIG. 115.—Buckets on one of the wheels of a 500-K.W. Curtis turbine.

The lower end of the shaft is supported by a cast-iron step-bearing, which takes the weight of the turbine and generator. This bearing is kept continually supplied with lubricating-oil under pressure, which is maintained by a small electric pump, mounted on the base of the turbine. An accumulator

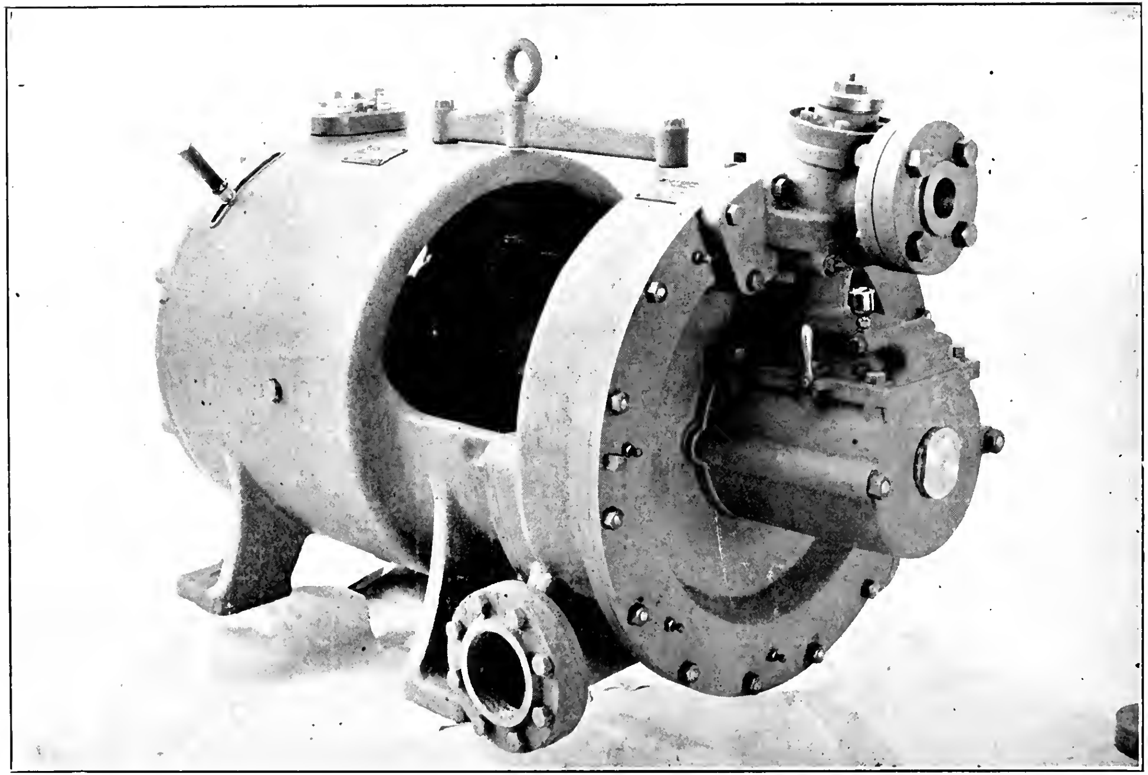


Fig. 116.—25-K.W. Curtis turbine, 3600 R.P.M., 125-volt generator.

is arranged, so that if the pump should break down, the supply of oil would be automatically continued.

When it is desired to run the turbine non-condensing, the exhaust is carried away from the first stage of the turbine through an atmospheric vent-pipe, fitted with an automatic

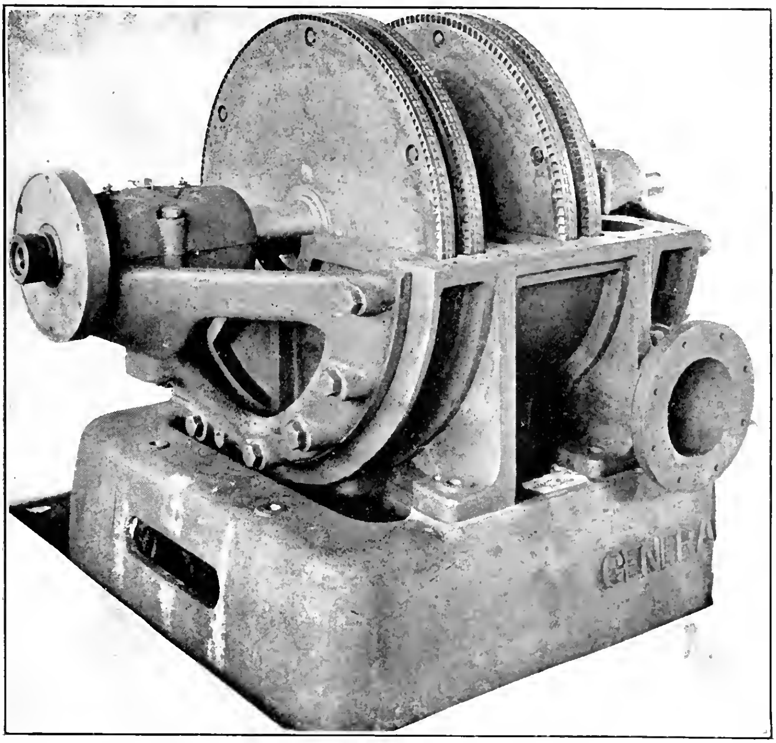


FIG. 117.—75-K.W. Curtis turbine-wheels, assembled in wheel-casing. Upper half of casing removed.

relief-valve. The second-stage nozzles may be shut off by a valve, when the turbine is to operate non-condensing.

In the supply-pipe is an automatically operated butterfly valve, arranged to cut off the steam-supply in case the speed of rotation becomes too high. A strainer is located between the throttle-valve and the steam-chest, to prevent

the entrance of any solid matter that might injure the working parts of the turbine.

The table of results of Curtis turbine tests shows the economy attained with the use of two-stage turbines at the

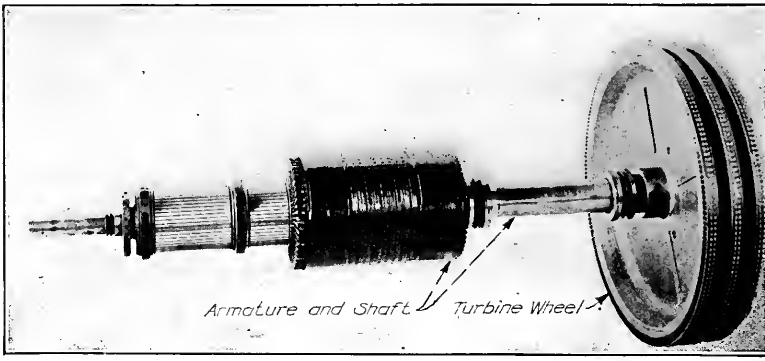
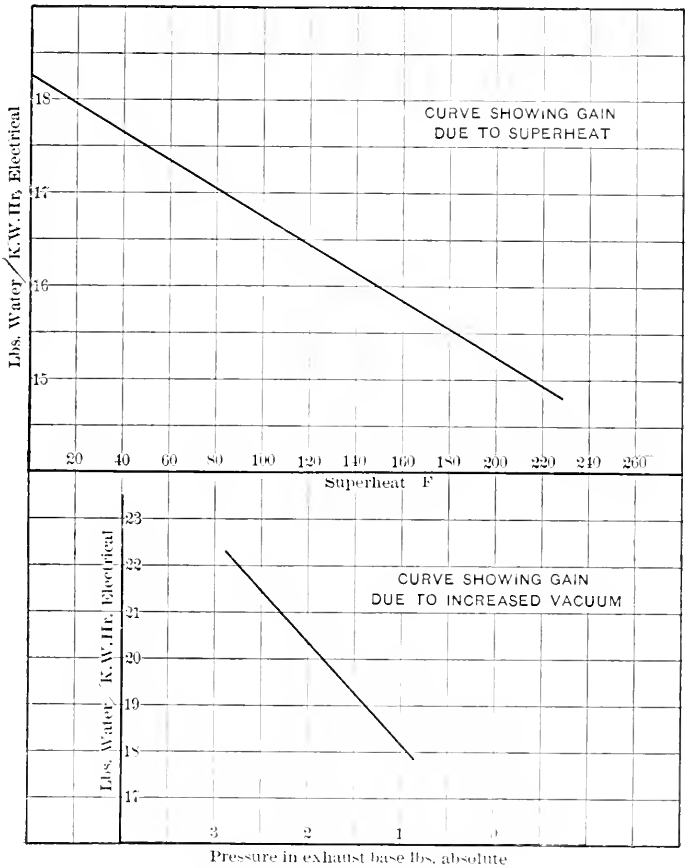



FIG. 118.—Rotating parts of 25-K.W., 3600 R.P.M., Curtis turbine, non-condensing.

Newport Station of the Old Colony Street Railway Company (see pages 282–3). The tests were made by Mr. George H. Barrus. Upon the basis of these results he makes the following comparison between the economy of the turbine and that of the direct-connected reciprocating steam-engine.

Taking the efficiency of the engine installation as 85 per cent, that is, Elec. H.P. ÷ I.H.P. = 0.85, for high-class compound steam-engines the consumption of dry steam may be taken as 13 ÷ 0.85 = 15.3 pounds per E.H.P. hour. The turbines tested, at full load, consumed 14.7 pounds per E.H.P. Thus the turbine was 4 per cent more economical at full load than a first-class compound reciprocating-engine, direct-connected. At half load the reciprocating-engine consumes 14.5 pounds per I.H.P. hour. The efficiency of the generator at half-load is 0.70, or the steam consumption is 14.5 ÷ 0.70 = 20.7 pounds per E.H.P. hour. The turbine consumed 15.9 pounds per E.H.P. hour; or, effected a gain of 23 per cent.




SUPERHEAT AND VACUUM CURVES
OF A
2000 K.W. CURTIS STEAM TURBINE

FIG. 119.

TESTS OF 500-K.W. CURTIS TURBINE, AT NEWPORT STATION OF THE OLD COLONY RAILWAY CO.

| | Full Load. | | | | | Overload. | | |
|---|---|---|---|---|---|---|---|---|
| | Saturated Steam. | Superheated Steam. | | | | Superheated Steam. 25 % Overload. | Saturated Steam. 50 % Overload. | Saturated Steam. 50% Overload. |
| 1. Duration, hours | 3.0 | 3.0 | 2.5 | 3.0 | 2.0 | 2.0 | 2.0 | 1.75 |
| 2. Moisture in steam, per cent | 1.9 | ........ | ........ | ........ | ........ | ........ | 1.3 | 1.6 |
| 3. Dry steam used per hour, lbs. | 10470 | 10558 | 10000.3 | 9411.7 | 8269.7 | 11950 | 15453 | 17260 |
| 4. K.W. used by condenser auxiliaries | 12.3 | 12.6 | 12.6 | 15.1 | 13.1 | 12.7 | 12.8 | 13.9 |
| 5. K.W. output, total | 529.4 | 539.5 | 512.3 | 528.9 | 519.7 | 665.7 | 764.3 | 780.3 |
| 6. Electrical H.P. output, total | 709.4 | 722.9 | 686.5 | 708.7 | 696.4 | 892.0 | 1024.2 | 1045.6 |
| 7. Steam-pressure near throttle, pounds by gage | 141.6 | 144.0 | 145.4 | 147.1 | 145.7 | 142.5 | 146.2 | 146.6 |
| 8. Vacuum in first stage, inches mercury | 8.3–9.9 | 8.4–10.3 | 9.6–10.7 | 9.8–10.4 | 11.3–13.1 | 5.1 | +0.7 pd.—3″ | +1 pd.—1″ |
| 9. Vacuum in second stage, and exhaust-chamber, inches mercury | 28.6 | 28.7 | 29.1 | 28.5 | 29.6 | 28.8 | 28.7 | 26.6 |
| 10. Absolute pressure in exhaust-chamber, inches | 1.0 | ........ | 1.0 | 1.1 | 0.9 | ........ | 1.4 | 3.0 |
| 11. Barometer, inches | 29.7 | 29.9 | 30.0 | 29.6 | 29.8 | 30.1 | 30.0 | 29.6 |
| 12. Temperature of steam at superheater, degrees F. | ........ | ........ | 450.5 | 537.0 | 695.8 | | | |
| 13 Temperature of steam at throttle, degrees F. | 361.1 | 377.3 | 423.1 | 512.3 | 651.0 | 485.1 | 361.4 | 361.6 |
| 14 Superheat at throttle, deg. F. | ........ | 16.4 | 61.8 | 150.5 | 289.6 | 121.9 | | |
| 15 Temperature in first stage, " | 194.4 | 194.0 | 191.5 | 253.4 | 337.1 | 251.5 | 215.0 | 220.0 |
| 16 Superheat in first stage, " | ........ | 0.0 | 0.0 | 62.4 | 150.0 | 47.5 | | |
| 17. Temperature in exhaust passage, degrees F | 78.0 | 78.7 | 76.9 | 79.8 | 99.0 | 81.5 | 87.3 | 117.0 |
| 18. Superheat in exhaust-passage, degrees F | ........ | 0.0 | 0.0 | 0.0 | 21.0 | 0.0 | | |
| 19 Average revolutions per minute | 1812.0 | 1829.0 | 1830.0 | 1835.0 | 1850.0 | 1815.0 | 1814.0 | 1807.0 |
| 20 Dry steam per K.W. hour, lbs. | 19.78 | 19.57 | 19.52 | 17.79 | 15.91 | 17.95 | 20.22 | 22.12 |
| 21. Dry steam per E.H.P. hour, lbs. | 14.76 | 14.61 | 14.57 | 13.28 | 11.87 | 13.40 | 15.09 | 16.51 |
| 22. B.T.U. per E.H.P. hour | 16923 | 16855 | 17076 | 16167 | 15012 | 16119 | 17165 | 18291 |

TESTS OF 500-K.W. CURTIS TURBINE, AT NEWPORT STATION OF THE OLD COLONY RAILWAY CO.—*Continued.*

| | Three-quarter Load. | | | | Half Load. | Quarter Load. | | No Load. | Commercial Load of St. tion. | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Saturated Steam. | | | Super-heated Steam. | Saturated Steam. | Saturated Steam. | Super-heated Steam. | Saturated Steam. | And about 200 K.W. Rheostat Load. Saturated Steam. | Station Load Only. Saturated Steam. |
| 1. Duration, hours | 2.0 | 4.0 | 1.5 | 3.0 | 4.0 | 3.0 | 1.0 | 3.5 | 12.22 | 15.0 |
| 2. Moisture in steam, per cent | 1.2 | 0.9 | 1.2 | ...... | 0.7 | 1.0 | ...... | 0.9 | 3.0 | 2.1 |
| 3. Dry steam used per hour, lbs. | 7022 | 8178 | 8299 | 6251 | 5711.5 | 3896.8 | 3224 | 1408.5 | | |
| 4. K.W. used by condenser auxiliaries | 15.6 | 12.6 | 12.7 | 16.0 | 13.0 | 15.3 | 13.0 | 15.4 | 14.9 | 18.5 |
| 5. K.W. output, total | 325.2 | 389.6 | 401.0 | 331.5 | 267.1 | 139.9 | 137.2 | 15.4 | 421.9 | 253.2 |
| 6. Electrical H.P. output, total | 435.8 | 522.1 | 537.3 | 444.2 | 357.9 | 187.5 | 183.8 | 20.6 | 565.3 | 339.3 |
| 7. Steam-pressure near throttle, pounds by gage | 145.8 | 147.7 | 147.9 | 146.4 | 145.5 | 147.5 | 147.8 | 148.0 | 146.8 | 146.3 |
| 8. Vacuum in first stage, inches mercury | 15.8–16.3 | 14–16 | 13.6–15.1 | 17–18.2 | 18–21 | 23–24 | 23.8 | 27.8–28.7 | 12–11 | 18.9–21.1 |
| 9. Vacuum in second stage and exhaust-chamber, in. mercury | 28.4 | 29.0 | 28.8 | 28.4 | 28.6 | 28.8 | 28.5 | 28.7 | 29.0 | 29.3 |
| 10. Absolute pressure in exhaust-chamber, inches | 1.1 | 1.1 | 1.0 | 1.0 | 0.8 | 1.3 | 1.0 | 1.4 | 1.0 | 0.7 |
| 11. Barometer, inches | 29.6 | 30.2 | 29.8 | 29.5 | 29.5 | 30.1 | 29.7 | 30.1 | 30.0 | 30.1 |
| 12. Temperature of steam at superheater, degrees F | ...... | ...... | ...... | 539.0 | ...... | ...... | 541.0 | | | |
| 13. Temperature of steam at throttle, degrees F | 361.2 | 362.0 | 362.4 | 504.8 | 362.3 | 361.9 | 505.0 | 362.1 | | |
| 14. Superheat at throttle, deg. F. | ...... | ...... | ...... | 143.2 | ...... | ...... | 142.9 | | | |
| 15. Temperature in first stage " | 174.7 | 182.2 | 182.2 | 208.0 | 165.3 | 148.9 | 216.0 | 117.8 | 184.0 | 163.3 |
| 16. Superheat in first stage, " | ...... | ...... | ...... | 38.0 | ...... | ...... | 74.0 | | | |
| 17. Temperature in exhaust-passage, degrees F | 81.3 | 82.6 | 77.1 | 78.6 | 71.6 | 86.9 | 96.0 | 101–116 | 79.3 | 74.2 |
| 18. Superheat in exhaust-passage, degrees F | ...... | ...... | ...... | 0.0 | ...... | ...... | 16.0 | | | |
| 19. Average revolutions per minute | 1841.0 | 1818.0 | 1822.0 | 1860.0 | 1823.5 | 1821.0 | 1875.0 | 1826.0 | 1838.0 | 1852.0 |
| 20. Dry steam per K.W. hour, lbs. | 21.59 | 20.99 | 20.69 | 18.86 | 21.38 | 27.85 | 23.49 | 91.5 | Turbine alone 20.70<br>Whole plant 24.31 | 22.38<br>26.68 |
| 21. Dry steam per E.H.P. hour, lbs. | 16.11 | 15.66 | 15.44 | 14.07 | 15.95 | 20.78 | 17.54 | 88.4 | Turbine alone 15.15<br>Whole plant 18.15 | 16.70<br>19.91 |
| 22. B.T.U. per E.H.P. hour | 18422 | 17892 | 17725 | 17095 | 18393 | 23654 | 21008 | 76895 | | |

Continuing, Mr. Barrus says: "The coal consumption on Jan. 15 was 2.54 pounds dry coal per K.W. hour of total output. If this test had been made with furnace efficiency as high as has been obtained with these boilers, the figure would have been 2.29 pounds of coal. There was an abnormal loss of steam between boilers and turbine, being 14.8 per cent and 16.1 per cent. In good practice this should not be over 7.5 per cent. Allowing for such a loss, the coal consumption would be 2.12 pounds per K.W. hour, or 1.58 per E.H.P. hour. Compared with power-station practice, this figure should be converted to switchboard output, and coal slightly wet. Allowing for current used by condenser auxiliaries, as 14.9 K.W., and for 4 per cent moisture in coal, the consumption of wet coalper K.W. hour of switchboard output, in good practice under these circumstances (the average net load being 407 K.W.), becomes 2.29 pounds. With corresponding high-class reciprocating-engine stations, the coal consumption per K.W. hour, of switch-board output, is from 2.5 to 2.6 pounds.

"These tests were made with two-stage turbines, and further economy may be expected from turbines with a larger number of stages.

"The advantage of superheating revealed by the Newport tests, on coal basis, is only 4.4 per cent under the most favorable conditions of temperature and efficiency. This result was obtained with a temperature of 700° at the superheater. There is good reason for expecting that increasing the number of stages of the turbine will be attended by a proportional gain, due to superheating, over the two-stage machine. Whatever percentage of saving in steam consumption may thus be secured, there will be the same percentage of increase in coal economy, and the improvement will be clear gain."

### Economy of Turbine expressed in Heat-units per Electrical Horse-power.

The B.T.U. per E.H.P. hour were 16,923, at full load, with saturated steam, and 15,012 with 289.6° superheat (using 0.48 as the specific heat of superheated steam). The heat utilized in evaporation per pound of dry coal was 10,765 B.T.U. On this basis the above figures represent a consumption of 1.57 pounds dry coal per E. H. P. hour for saturated steam, and 1.39 pounds for superheated steam per E.H.P. hour. The heat consumptions given are equivalent to 282 B.T.U. per E.H.P. per minute for saturated steam, and 250 for superheated steam.

The comparisons given above, between the performance of turbines and compound reciprocating-engines are based upon the results of one particular type of turbine, because the figures were at hand, but any of the well-developed types would give approximately the same results under similar conditions.

The turbine, although possessing distinct advantages in point of convenience, space, oil and attendance required, has not yet equaled the steam economy attained with the best triple-expansion stationary reciprocating engines. A comprehensive comparison places the two types of motor very close together in general utility and effectiveness, with the turbine gaining ground for power station service because of its simplicity.

Fig. 113 shows one of the latest designs of Curtis turbine, having four stages and rated capacity 2000 K.W. The results in the following table are from a test made at Schenectady in

| | Full Load. | Half Load. | Quarter Load. | No Load. |
|---|---|---|---|---|
| Duration of test, hours | 1.25 | 0.916 | 1.00 | 1.33 |
| Steam-pressure, gage | 166.3 | 170.2 | 155.5 | 154.5 |
| Back pressure, inches mercury | 1.49 | 1.40 | 1.45 | 1.85 |
| Superheat, degrees F. | 207 | 120 | 204 | 153 |
| Load in kilowatts | 2023.7 | 1066.7 | 555 | 1510.5* |
| Steam per kilowatt hour, pounds | 15.02 | 16.31 | 18.09 | |

* Total water per hour.

1905, under the direction of Messrs. Sargent and Lundy of Chicago. The revolutions per minute were 900.

In Figs. 116 to 118 are shown small horizontal Curtis turbines, direct-connected to generators. The latter are direct-current machines and operate at the speeds of revolution given in the table on page 287.

**Other types of turbine** are about to be introduced in this country, similar to the output of European firms. The Hooven-Owens-Rentschler Company of Hamilton, Ohio, is building the Hamilton-Holzwarth turbine, which is of the general character of the Rateau turbine, operating upon the impulse principle entirely, and having several compartments, each containing a rotating wheel.

The Zoelly turbine, also of the many-stage impulse type, is being manufactured by the Providence Engineering Works, of Providence, R. I.

**Capacity and Speed of Revolution of Turbines.**—The following tables give particulars of Parsons and of Curtis turbines, as built for operating electric generators.

PARSONS TURBINES.

| K.W. | R.P.M. 60-cycle. | R.P.M. 25-cycle. |
|---|---|---|
| 300 | 3600 | 1500 |
| 400 | 3000 | .... |
| 500 | 3600 | 1500 |
| 750 | 1800 | 1500 |
| 1000 | 1800 | 1500 |
| 1500 | 1200 | 1500 |
| 2000 | 1200 | 1500 |
| 3500 | 720 | 750 |
| 5000 | 720 | 750 |
| 6000 | 720 | 750 |
| 7500 | 720 | 750 |

200 K.W. direct-current, 1850 R.P.M.

The speed of revolution of De Laval turbine generators is given in the tables of tests. The speed of revolution of the turbine-wheel is usually ten times that of the generator armature.

## CURTIS TURBINES.

### DIRECT-CURRENT.

Horizontal Shaft.

| Class. | Poles. | K.W. | R.P.M. | Volts. |
|---|---|---|---|---|
| C | 2 | 15 | 4000 | 80–125 |
| " | 2 | 25 | 3600 | 125–250 |
| " | 4 | 75 | 2400 | 125–250 |
| " | 4 | 150 | 2000 | 125–250 |
| " | 4 | 300 | 1500 | 125–550 |

Vertical Shaft.

| Class. | Poles. | K.W. | R.P.M. | Volts. |
|---|---|---|---|---|
| C | 4 | 500 | 1800 | 550 |

### ALTERNATING-CURRENT.

Vertical Shaft—60-cycle.

| Class. | Poles. | K.W. | R.P.M. | Volts. |
|---|---|---|---|---|
| ATB | 4 | 300 | 1800 | 240– 4000 |
| " | 4 | 500 | 1800 | 240– 6600 |
| " | 6 | 1000 | 1200 | 480– 6600 |
| " | 8 | 1500 | 900 | 480– 6600 |
| " | 8 | 2000 | 900 | 1150–13200 |
| " | 12 | 3000 | 600 | 600–13200 |
| " | 10 | 5000 | 720 | 2300–13200 |

25-cycle.

| Class. | Poles. | K.W. | R.P.M. | Volts. |
|---|---|---|---|---|
| ATB | 2 | 300 | 1500 | 370– 6600 |
| " | 2 | 800 | 1500 | 600–13200 |
| " | 4 | 2000 | 750 | 2300–13200 |
| " | 4 | 5000 | 750 | 2300–13200 |

**Clearances in Turbines.**—In impulse-turbines fitted with guide-buckets the clearance between buckets is important; but, as was shown in the experimental work described in Chapter VI, small clearances, such as are necessary for mechanical operation of the wheels, do not seriously affect the efficiency. The following clearances are recommended by the General Electric Company.*

| Turbine. | | Clearances. | | | |
|---|---|---|---|---|---|
| Rating. | Stages. | First Stage. | Second Stage. | Third Stage. | Fourth Stage. |
| 500 | 4 | 0.06 inch | 0.06 inch | 0.06 inch | 0.06 inch |
| 800 | 4 | .07 " | .07 " | .07 " | .07 " |
| 1000 | 7 | .08 " | .08 " | .08 " | .15 " |
| 1500 | 4 | .06 " | .06 " | .06 " | .08 " |
| 2000 | 4 | .06 " | .06 " | .08 " | .08 " |
| 3000 | 4 | .07 " | .07 " | .07 " | .08 " |
| 5000 | 4 | .07 " | .07 " | .07 " | .08 " |
| 5000 | 6 | .10 " | .1 " | .1 " | .2 " |

In the ideal many-stage turbine, since there is no drop in pressure in any given stage after the steam leaves the nozzles, the direction of flow is determined by the nozzles and guide-buckets, and the clearance past the periphery of the wheels is of little or no consequence. A certain amount of clearance is desirable from mechanical considerations, and this apparently does not interfere with the efficiency of the actual machine.

In the reaction type of turbine it is the limitation of clearance past the periphery of the blades that is important, and not that between the rows of blades. This is because there is expansion of the steam all along the turbine, and the steam tends to flow in all directions. Leakage past the ends of the blades is, therefore, to be prevented, and the clearances are kept as small as possible. Knowledge regarding the expansion of the spindle and casing caused by the temperatures attained in operation is possessed by turbine-builders, and the clearances are arranged accordingly. The clearances between rows

* See Report of Committee for the Investigation of the Steam-turbine, National Electric Light Assoc., June, 1905.

of blades vary from $\frac{1}{8}$ or $\frac{3}{16}$ inch in the high-pressure stages to one inch or even more in the lower-pressure stages. The clearance between the tips of the blades and the casing or spindle, as the case may be, is limited to a few one-thousandths or a few one-hundredths of an inch, according to circumstances.

**The gain due to increase of vacuum** is illustrated by the following extract from the "Report of the Committee for the Investigation of the Steam-turbine," appointed by the National Electric Light Assoc., and before referred to:

"From a recent test made by your committee on a 2000-K.W. turbine, different vacua were run for the specific purpose of obtaining the vacuum effect; it was found that for this turbine running at 1800 kilowatts the increase in economy is 5.2 per cent from 25-inch to 27-inch vacuum, and 6.75 per cent from 27-inch to 28-inch.

"Under the following assumed conditions the economy effected in operating under high vacuum would work out somewhat as follows:

| | | |
|---|---|---|
| Assumed size of unit, K.W. | 2000 | |
| Average load | 1500 | |
| Hours run per day | 15 | |
| Hours run per year (300 days) | 4500 | |
| Price coal per ton, 2000 pounds | $3.00 | |
| Evaporation | 9 pounds | |
| Economy pounds water per kilowatt | 22 | |
| Rise in vacuum | 26–28 inches | |
| Assumed per cent increase of economy due to increase of vacuum from 26–28 inches | 6 per cent | |
| Water saved per K.W.-hour | 1.32 | |
| Water saved per year | 140,000 cu. ft. | |
| Cost of water saved per year at 2.5$ | $35.00 | $35.00 |
| Coal saved per year | 500 tons | |
| Cost of coal saved per year at $3.00 | $1500.00 | $1500.00 |
| | | $1535.00 |
| Increased cost of condenser plant for 28-inch over that of 26-inch assumed $5000.00; interest on above at 5 per cent, depreciation 10 per cent, other fixed charges, including repairs, 2 per cent, total 17 per cent | $850.00 | 850.00 |
| Saving per year | | $685.00 |

| | | |
|---|---|---|
| Saving per year............................ | | $685.00 |
| The above does not include the extra cost in steam to run the larger auxiliaries, but, inasmuch as such exhaust-steam would return a benefit to the feed-water if they were all steam-driven, we will assume that the extra cost in water is 2 per cent of the total steam guaranteed by the turbine and will amount per year to.................. | $12.00 | 12.00 |
| Total net saving........................ | | $673.00 |
| With interest at 5 per cent this represents a capital saving of................................ | | 13,460.00 |

**Sizes of Condensers and Auxiliaries.**—"The turbine installations concerning which we have received information, where 28 inches of vacuum is maintained with a cooling-water temperature of 70 degrees F., show a miniumm ratio of cooling surface in the condenser to steam condensed, per minute, of 6.9 square feet per pound. But the more usual ratio, even where the cooling water is from 5 to 10 degrees lower in temperature, is 8 to 9 square feet per pound. In the first instance noted above it is to be remarked that the ratio of circulating water to condensed steam is 70 to 1. With greater cooling surface ratios the proportion of cooling water is reduced.

"In actual practice, for temperatures of cooling water ranging from 60 to 70 degrees, circulating-pumps have been installed for volumes of cooling water ranging from 40 to 70 times that of the water of condensation. At the low ratio of 40 to 1 the cooling water temperature must be close to 60 degrees for so high a vacuum as 27.5 inches, and even then considerable difficulty is experienced in maintaining the 27.5 inches, unless the ratio of cooling surface to pounds of steam condensed per minute is 8 to 1.

**Steam Used by Auxiliaries.**—"These figures are obtained from letters sent to us by turbine owners:

3 200-K.W. De Laval exhausting into one condenser.

3000 gallons per minute circulating-pump; 2-stage dry-vacuum pumps $8\times12\times\frac{12}{16}$; duplex wet-vacuum pump; 15-K.W. turbine exciter. Steam by auxiliaries, 2.6 pounds per kilowatt.

Byllesby & Co.:

Steam per kilowatt at half load, 3.5 pounds.

BOSTON EDISON COMPANY.

5000-K.W. Turbine Unit.

| | | | |
|---|---|---|---|
| Kilowatts on turbine | 2713 | 3410 | 4758 |
| Vacuum | 28.4 | 28.7 | 28.6 |
| Barometer | 29.53 | 29.95 | 29.96 |
| Boiler-feed pump, I.H.P. | 13.9 | 23.7 | 27.4 |
| Circulating-pump, I.H.P. | 69.1 | 69.1 | 69.1 |
| Dry-vacuum pump, I.H.P. | 24.3 | 23.2 | 23.8 |
| Step-bearing pump, I.H.P. | 6.4 | 5.8 | 5.6 |
| Wet-vacuum pump, E.H.P. | 8.6 | 9.2 | 9.8 |
| Total power for auxiliaries | 122.3 | 131 | 135.7 |
| Per cent of power of auxiliaries to power of turbine | 3.4 | 2.9 | 2.1 |
| Per cent of water used by auxiliaries to that used by turbine | 8.4 | 7.4 | 5.7 |

TEST REPORTED BY NASHUA LIGHT, HEAT, AND POWER COMPANY.

500-K.W. Curtis, Rated Water per Hour 20.5 Pounds.

| | Steam per Hour, Saturated. | Steam per Hour, Superheat. | Pounds Difference per Hour. | One Per Cent Difference. | Degrees Superheat. | |
|---|---|---|---|---|---|---|
| Accumulator-pump | 130.9 | 130.9 | | | | |
| Dry-air pump | 181.58 | 183.13 | | | | |
| Boiler-feed pump | 352.15 | 249.58 | 102.57 | 29.12 | 71.98 | Feed pumps act as wet-vacuum pumps. |
| Westinghouse jun. driving circ. pumps | 663.64 | 439.36 | 224.28 | 33.79 | 97.65 | |
| Totals | 1328.27 | 1002.97 | | | | |

Per cent of rated water consumption of turbines at full load .......... 12.9 per cent, 9.78 per cent
Dry-air pump, 6″ and 12″ by 12″ stroke, 93 R.P.M.
Boiler-feed pumps, 7.5″ and 4.5″ by 10″ stroke, 98 R.P.M.
Centrifugal-pump engine, 7″ by 6″ stroke.

"It is, however, a question whether the extra cost of steam for driving larger auxiliaries for high vacuum work is of any great moment, as such steam is of considerable value in the feed-heater. It is to be noted also that these figures are for total consumption of auxiliaries, and that the increase of steam

necessary to obtain two inches more than 26 or 28 inches must necessarily be very small.

"An important feature of operation with high vacuum is the necessity of having air-tight stuffing-boxes and pipe-joints, lack of which results in loss of economy to the turbine, and increased consumption of steam by the dry-vacuum pumps and circulating pumps.

"Undoubtedly the best arrangement of the condensing plant is the use of a counter-current condenser, placed as close to the exhaust-nozzle as possible and with the dry-air pumps drawing from the condenser at the point of coolest circulating water; this pump also so placed that the minimum of pipe connection can be used. With this arrangement the possibility of air-leaks would be greatly reduced, the quantity of circulating water would also be lessened, owing to the lower tension of the air which has just left the coldest tubes of the condenser. We believe that it is important, in lowering operating costs, that the above design of the installation should in all cases be followed as rigidly as individual conditions will permit.

"From the experience obtained in their own plants and in testing others, the committee recommends that the capacity in cubic feet of volume swept by the air-piston of the dry-air pump be not less than 45 times the volume of the condensed steam; and where overload conditions are frequent, not less than 50 times the water (condensed steam) volume."

**General Remarks on Steam-turbine Design.**—The experimental work on buckets, discussed in Chapter VI, indicates that the placing of a number of rows of moving and stationary buckets in a single stage of an impulse-turbine may lead to an accumulation, or backing up, of pressure. This may be caused by any of the following conditions:

(*a*) Insufficient area for the passage of steam, especially in the last wheels of the stage.

(*b*) Discharge side of the buckets making too small an angle with the direction of motion of the buckets.

(*c*) Bucket surfaces opposing undue frictional resistance to the passage of steam.

(*d*) The steam-passages from one wheel to another being indirect and opposing undue obstruction to the flow of steam. If, in order to reach a succeeding row of buckets, the steam has to traverse the surface of a rotating wheel, this may interfere with free flow and cause loss.

These conditions may prevent the production of the desired rotative effort in the stage in question, and thus call for modifications in the area and character of steam-passages, in the bucket exit angles, and, assuming it to be practicable, in the degree of smoothness to which the bucket surfaces are finished.

In one of the most recent types of Curtis turbine there are four stages, and one rotating disk or wheel in each stage, carrying two rows of buckets. The 2000-K.W. turbine shown in Fig. 113 is of this type.

In general, as great freedom as possible is required for passage of the steam through the high-pressure stages of the turbine. But, at the same time, sufficient area of buckets must be provided for the steam to act against, and this may call for an increased number of buckets in the last wheels of a stage, as the exit angles are increased.

In the Parsons type freedom of steam-passage is equally desirable, and in general the requirements are similar to those just stated. It is desirable to keep the steam velocities low, and, while certain undesirable features appear, it is quite possible to design a reaction-turbine having practically uniform steam velocities throughout the machine.

*In conclusion* it should be said that the determination of sizes and general proportions of mechanical devices of all kinds, and more especially in cases of departure from the beaten path such as that now being made by the builders of steam-turbines of the various types, is only a first step towards bringing forth satisfactory results as viewed from an engineering standpoint. The development of satisfactory details and the commercially successful production of the finished machine call for technical

and mechanical skill combined with business ability all of the highest order, and unstinted credit is due to the men who have worked and are working to perfect the mechanism of the steam-turbine.

**Note regarding the Design of Condensers and Air-pumps.**—In a paper presented to the Inst. of Naval Architects, London, Apr. 1906 (see reprint, "Engineering," Apr. 13–20), Prof. R. L. Weighton describes very complete experimental work performed in order to ascertain the relative efficiencies of the surface condenser as ordinarily built for both stationary and marine work, and the surface condenser to which the name "Contraflo" has been given. The conclusions are of exceptional interest, and indicate that condensers and air-pumps are commonly made of considerably greater size and capacity than would be found either necessary or desirable if the principles brought out in the paper were made use of in the design of those parts.

The type of counter-current condenser referred to on page 292 is a horizontal surface condenser, in which the cooling water and the exhaust steam enter in opposite directions, preferably with the steam entering at the bottom of the shell, and the water through the tubes at the top. The dry air-pump is then caused to draw from a connection at or near the top of the condenser shell.

## CHAPTER X.

### THE MARINE STEAM-TURBINE.

THE recent decision of the Cunard Steamship Company, and that of the British Admiralty, to install turbines in place of reciprocating-engines in various large and important vessels, have brought the marine steam-turbine very prominently before the public. This departure, made by conservative engineers who had access to all the existing data on the subject, has apparently been justified by the subsequent good behavior of the turbines already installed. The question as to the efficiency of the marine turbine must rest upon the results of tests of different classes of vessels under various conditions, but the trials made thus far are very gratifying in their results, and cover a fairly wide range of vessels, from the first small boat, Turbinia, of 32 knots speed, to the ocean liner Carmania, of about 19 knots speed, which has just completed her initial voyage successfully; and including the third-class cruiser Amethyst, in which the economy of the turbine, at the highest powers, exceeded that of the reciprocating-engine by as much as 40 per cent, and excelled in efficiency at all speeds above 14 knots per hour.

The following table gives particulars of practically all of the vessels which have been equipped with turbine machinery.

TURBINE STEAMERS—

(The table is from a paper by Mr. E. M. Speakman,

| Ref. No. | Date. | Vessel. | Service. | Owner. | Builder. |
|---|---|---|---|---|---|
| 1 | 1894* | Turbinia ....... | Experimental ... | C. A. Parsons........ | C. A. Parsons.... |
| 2 | 1900 | King Edward.... | Pleasure Steamer. | Turbine Steamers, Ltd. | Denny Bros. .... |
| 3 | 1901 | Queen Alexandra. | do....... | do......... | do........... |
| 4 | 1898 | Viper.......... | T. B. D......... | R. N................ | Hawthorn, Leslie & Co. |
| 5 | do | Cobra.......... | do........... | do................ | Armstrong, Whitworth & Co. |
| 6 | 1903 | Velox.......... | do........... | do................ | Hawthorn, Leslie & Co. |
| 7 | 1904 | Eden........... | do........... | do................ | do....... |
| 8 | 1905 | Coastal Destroyers.. | do........... | do................ | Thornycroft, Yarrow and White. |
| 9 | do | Ocean-going Destroyers. | do........... | do................ | Laird, Thornycroft, Armstrong, White, Hawthorn, and Leslie & Co. |
| 10 | do. | Experimental Destroyers. | do........... | do................ | ............... |
| 11 | 1903 | Tarantula....... | S. Y........... | W. K. Vanderbilt..... | Yarrow......... |
| 12 | do. | Lorena......... | do............ | A. L. Barbour........ | Ramage & Ferguson. |
| 13 | do. | Emerald........ | do............ | Sir C. Furness........ | Stephen & Sons.. |
| 14 | 1905 | Albion ......... | do............ | Sir G. Newnes........ | Swan & Hunter.. |
| 15 | do | Narcissus....... | do............ | A. E. Mundy. ........ | Fairfield. ....... |
| 16 | do. | Royal Yacht.... | do............ | H. M. King Edward. .. | A. & J. Inglis.... |
| 17 | do | Mahroussah. .... | do............ | The Khedive of Egypt. | do. (rebuilding). . |
| 18 | 1903 | The Queen...... | Channel Steamer | S. E. & Chatham Ry.Co. | Denny Bros..... |
| 19 | do. | Brighton........ | do....... | L. B. & South-Coast Ry. Co. | do........... |
| 20 | 1904 | Princess Maud... | do....... | Stranraer & Larne Service. | do........... |
| 21 | do. | Londonderry. . . | do....... | Midland Railway Co... | do........... |
| 22 | do. | Manxman....... | do....... | do.............. | Vickers, Sons & Maxim. |
| 23 | 1905 | Viking.......... | do....... | Isle of Man S. S. Co. ... | Armstrong, Whitworth & Co. |
| 24 | do | Onward......... | do....... | S. E. & Chatham Ry. Co. | Denny Bros. .... |
| 25 | do | Dieppe......... | do....... | L. B. & S.-Coast Ry. Co. | Fairfield. ....... |
| 26 | do | ................ | do..... . | G. & J. Burns........ | do.......... |
| 27 | do | ................ | do....... | Great Western Ry. Co. | J. Brown & Co., and Laird & Co. |
| 28 | do. | Princess Elizabeth | do....... | Belgian Government... | Cockerill....... |
| 29 | do | Kaiser.......... | do....... | Hamburg - Heligoland S. S. Co. | Vulcan Co....... |

* Rebuilt 1896.

Remarks.—1 Only one screw, 28″ diameter, now fitted to each shaft.
2. Put in service July, 1901.
3. Put in service July, 1902. Very largely used for experimental trials.
4. Launched 6/9/99. Ran ashore and lost during naval manœuvres in 1901. Trials made in 1900.
5 Sank at sea in September, 1901.
6 Reciprocating cruising engines on inner shafts, 7½″, 11″, and 16″×9″ stroke; 400 R.P.M. Launched 2/1902.
8 Twelve building.
9 Five building.
10 Details under consideration
11 One 3′ 0′ screw now fitted to each shaft.

GENERAL DIMENSIONS AND DATA.

Transactions of the Inst. of Engineers and Shipbuilders of Scotland, 1905.)

| Length. | Beam | Depth. | Draught. | Speed. | Equivalent I.H.P. | No. of Shafts. | Screws per Shaft. | R.P.M. | Boiler Pressure | Displacement. | Propeller Diameter. | Ref. No |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| ′ ″ | ′ ″ | ′ ″ | ′ ″ | Knots. | | | | | Lbs. | Tons. | ′ ″ | |
| 100 | 9 | ..... | 3 | 32.0 | 2,000 | 3 | 3 | 2,300 | 210 | 15 | 1 6 | 1 |
| 250 | 30 | 10 6 | 6 | 20.48 | 3,500 | 3 | 1 center,<br>2 wing | 505 c<br>750 w | 150 | 700 | 4 9<br>4 4 | 2 |
| 270 | 32 | 11 6 | 6 6 | 21.43 | 4,400 | 3 | 1 | 750 e<br>1,090w | 150 | 900 | ...... | 3 |
| 210 | 21 | 12 9 | 6 9 | 36.58 | 13,000 | 4 | 2 | 1,180 | 240 | 390 | 3 4 | 4 |
| 223 | 20 6 | 13 6 | 7 3 | 30.2 | 10,000 | 4 | 3 | 1,050 | 240 | 450 | 2 9 | 5 |
| 210 | 21 | 12 9 | 7 3 | 27.1 | 7,000 | 4 | 1 | 890 | 240 | 440 | 4 0 | 6 |
| 220 | 23 6 | 14 3 | 8 3 | 26.2 | 7,500 | 3 | 2 | 940 | 250 | 570 | 3 3 | 7 |
| 175 | ..... | ..... | ..... | 26.0 | 3,600 | 3 | 1 | 1,200 | 220 | 225 | 3 0 | 8 |
| 250 | ..... | ..... | ..... | 33.0 | 15,000 | 3 | 1 | 700 | 220 | 800 | 6 0 | 9 |
| 320 | ..... | ..... | ..... | 36.0 | 28,000 | 4 | 1 | 600 | 250 | 1,500 | 7 0 | 10 |
| 152 6 | 15 3 | 8 4 | 5 0 | 25.36 | 2,200 | 3 | 3 | 1,200 | 225 | 145 | ....... | 11 |
| 253 | 33 3 | 20 4 | 13 0 | 18.02 | 3,800 | 3 | 1 | 550 c<br>700 w | 180 | 1,400 | 4 8<br>4 0 | 12 |
| 198 | 28 7 | 18 6 | ..... | 15.0 | 1,400 | 3 | 1 | 900 | 150 | 900 | ....... | 13 |
| 270 | 34 | ... | ..... | 15.0 | 1,800 | 3 | 1 | ...... | 150 | 1,250 | ....... | 14 |
| 245 | 27 6 | 16 3 | ..... | 14.5 | 1,250 | 2 | 1 | 550 | 160 | 782 | ....... | 15 |
| 310 | ..... | ..... | ..... | 18.0 | 4,000 | 3 | 1 | ...... | ..... | 2,800 | ....... | 16 |
| 400 | 42 | 26 6 | .... | 18.0 | 6,500 | 3 | 1 | ...... | 150 | 3,100 | ....... | 17 |
| 310 | 40 | 25 | 10 6 | 21.73 | 8,500 | 3 | 1 | 480 c<br>500 w | 150 | ....... | 6 0<br>5 7 | 18 |
| 280 | 34 | 22 | 9 0 | 21.5 | 6,000 | 3 | 1 | 480 c<br>510 w | 150 | 1,200 | ....... | 19 |
| 300 | 40 | 24 6 | 19 6 | 20.7 | 6,500 | 3 | 1 | 600 | 150 | 1,750 | 5 0 | 20 |
| 330 | 42 | 25 6 | 10 6 | 22.3 | 7,000 | 3 | 1 | 670 c<br>750 w | 150 | 1,950 | 5 0 | 21 |
| 330 | 43 | 25 6 | 10 6 | 23.14 | 8,500 | 3 | 1 | 530 c<br>610 w | 200 | 2,000 | 6 2<br>5 7 | 22 |
| 350 | 42 | 17 3 | 10 6 | 23.53 | 9,500 | 3 | 1 | 430 | 160 | ....... | 6 6 | 23 |
| 310 | 40 | 25 | 10 6 | 22.9 | 8,000 | 3 | 1 | 540 | 150 | ....... | 6 0 | 24 |
| 280 | 34 8 | 14 6 | 9 3 | 21.75 | 6,500 | 3 | 1 | 600 | 150 | 1,360 | 5 3 | 25 |
| 340 | ..... | ..... | .... | ........ | 6,000 | 3 | 1 | 600 | ..... | ....... | ....... | 26 |
| 350 | 40 | ..... | 14 0 | 23.0 | 9,500 | 3 | 1 | 430 | 160 | ...... | ....... | 27 |
| 350 | 40 | ..... | 9 7 | 24.0 | 12,000 | 3 | 1 | 490 | 150 | 1,950 | ....... | 28 |
| 300 | 38 | ..... | 9 10 | 20.0 | 6,000 | 2 | 1 | 650 | ..... | 2,000 | ....... | 29 |

12. Yacht measurement.
14. Thames yacht measurement.
15. Thames yacht measurement. Only twin-screw Parsons installation.
17. In process of conversion from paddle engines to turbines. Vessel built in 1865 by Samuda.
18. Screws originally arranged as in King Edward. 13 knots astern speed.
20. Bow rudder fitted.
24. Sister ship Invicta.
25. See "Engineering," Aug. 18, 1905.
27. Three building.
28. Astern speed 16.0 knots; 415 R.P.M.
29. Curtis turbines.

TURBINE STEAMERS—GENERAL

| Ref. No. | Date. | Vessel. | Service. | Owner. | Builder. |
|---|---|---|---|---|---|
| 30 | 1904 | Lhassa. . . . . . . . . | Persian Gulf to India. Intermediate. | British India S. S. Co. . | Denny Bros. . . . . |
| 31 | do. | Loongana. . . . . . . | Inter-Colonial Service, Tasmania—Melbourne. | Union S. S. Co. of New Zealand. | do. . . . . . . . . . . . |
| 32 | do. | Turbinia II. . . . . | Pleasure Steamer Lake Ontario. | Turbine S. S. Co. . . . . . | Hawthorn, Leslie & Co. |
| 33 | 1905 | Maheno. . . . . . . . . | Inter-Colonial. . . | Union S. S. Co. of New Zealand. | Denny Bros. . . . . |
| 34 | do. | Bingera. . . . . . . . . | Australian Passenger. | . . . . . . . . . . . . . . . . . . . . . | Workman & Clarke. |
| 35 | do. | Victorian. . . . . . | Atlantic Intermediate Service | Allan S. S. Co. . . . . . . . . | do. . . . . . . . |
| 36 | do. | Carmania. . . . . . . | Atlantic Mail. . . . | Cunard Co. . . . . . . . . . . | John Brown & Co. |
| 37 | 1904 | Lusitania. . . . . . . Mauretania. . . . . | do. . . . . . . . . | do. . . . . . . . . . . . . . | J. Brown & Co., and Swan & Hunter. |
| 38 | do. | Amethyst. . . . . . . | 3d-class cruiser. . | R. N. . . . . . . . . . . . . . . . | Armstrong, Whitworth & Co. |
| 39 | 1905 | Lubeck. . . . . . . . . | do. . . . . . . | German Navy. . . . . . . . | Vulcan Co. . . . . . . |
| 40 | do. | Salem. . . . . . . . . . | Scout Cruiser. . . . | U. S. N. . . . . . . . . . . . . . | Bath Iron Works. |
| 41 | do. | Chester. . . . . . . . . | do. . . . . . . . . | do. . . . . . . . . . . . . . . . | Fore River S. & E. Co. |
| 42 | do. | Dreadnought . . . . | Battleship. . . . . | R. N. . . . . . . . . . . . . . . | Portsmouth Dockyard. |
| 43 | do. | Orion class. . . . . . | Armored Cruisers. | do. . . . . . . . . . . . . . . . | . . . . . . . . . . . . . . . . |
| 44 | | No. 243. . . . . . . . . | Experimental Torpedo Boat. | French Navy. . . . . . . . | Société des F. & C. Mediterranée. |
| 45 | | Libellule. . . . . . . . | do. . . . . . | do. . . . . . . . . . . . . . | . . . . . . . . . . . . . . . |
| 46 | 1903 | Caroline. . . . . . . . | do. . . . . . . | do. . . . . . . . . . . . . | Yarrow. . . . . . . . . . |
| 47 | 1904 | No. 293. . . . . . . . . | Torpedo Boat . . . | do. . . . . . . . . . . . . | Normand. . . . . . . |
| 48 | do. | No. 294. . . . . . . . . | do. . . . . . . . . | do. . . . . . . . . . . . . . | do. . . . . . . . . . . |
| 49 | 1905 | S. 125. . . . . . . . . . | T. B. D. . . . . . . . . | German Navy. . . . . . . | Schichau. . . . . . . . |
| 50 | 1903 | Revolution. . . . . . | Experimental S. Y. | Curtis Marine Turbine Co. | . . . . . . . . . . . . . . . . |

30. Sister ships Linka, Lunka, Lama.
35. Also Virginian, built by Stephen & Sons. Weight saved by adopting turbines, 400 tons. Passengers increased 60.
37. Two building.
38. See "Engineering," November 18, 1904.
41. Curtis turbines.
43. Designs still under consideration.

The principal reasons for the present tendency to adopt the steam-turbine in place of the reciprocating-engine for propelling ships of certain types are the following:

1. Decreased cost of operation as regards fuel, labor, oil, and repairs.

2. Vibration due to machinery is decreased.

3. Less weight of machinery and coal to be carried, resulting in greater speed.

4. Greater simplicity of machinery in construction and operation, causing less liability to accident and breakdown.

DIMENSIONS AND DATA—*(Continued)*.

| Length. | Beam | Depth. | Draught. | Speed. | Equivalent I.H.P. | No. of Shafts. | Screws per Shaft. | R.P.M. | Boiler Pressure. | Displacement. | Propeller Diameter. | Ref. No. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| ′ ″ | ′ ″ | ′ ″ | ′ ″ | Knots. | | | | | Lbs. | Tons. | ′ ″ | |
| 275 | 44 | 25 6 | ..... | 18.0 | 6,000 | 3 | 1 | ...... | 150 | 2,170 | ....... | 30 |
| 300 | 43 | 25 | 12 6 | 20.2 | 6,300 | 3 | 1 | 650 | 150 | 2,400 | 5 3 | 31 |
| 260 | 33 | 20 9 | 9 6 | 19.0 | 3,500 | 3 | 1 | 650 | 160 | 1,100 | 4 1½ | 32 |
| 400 | 50 | 33 6 | ..... | 17.5 | ...... | 3 | 1 | ...... | ..... | ....... | ....... | 33 |
| 300 | ..... | ..... | ..... | ........ | ...... | 3 | 1 | ...... | ..... | ....... | ....... | 34 |
| 540 | 60 | 42 6 | 27 6 | 19.5 | 12,000 | 3 | 1 | 275 | 180 | 13,000 | 8 9 | 35 |
| 678 | 72 | 52 | 32 | 21.0 | 21,000 | 3 | 1 | 185 | 195 | 30,000 | 14 0 | 36 |
| 785 | 88 | ..... | 33 6 | 25.0 | 68,000 | 4 | 1 | 165 | 195 | 38,000 | 17 3 | 37 |
| 360 | 40 | ..... | 14 6 | 21.75 trial<br>23.63 des. | 9,800<br>14,000 | 3 | 1 | 450 c<br>490 w | 250 | 3,000 | 6 6 | 38 |
| 341 | 43 3 | ..... | 16 6 | 22.0 des. | 10,000 | 4 | 1 | 650 | ..... | 3,200 | ....... | 39 |
| 420 | 46 8 | ..... | 16 9 | 24.0 | 16,000 | 4 | 1 | 500 | 250 | 3,750 | 6 6 | 40 |
| 420 | 46 8 | ..... | 16 9 | 24.0 | 16,000 | 2 | 1 | 350 | 250 | 3,750 | ....... | 41 |
| ..... | ..... | ..... | ..... | 21.0 | 23,000 | 4 | 1 | 300 | 250 | 18,000 | 9 3 | 42 |
| ..... | ..... | ..... | ..... | 24.0 | 28,000 | 4 | 1 | ...... | 250 | ....... | ....... | 43 |
| ..... | ..... | ..... | ..... | 21.0 | 1,800 | 2 | Various | 1,800 | ..... | 92 | Various | 44 |
| ..... | ..... | ..... | ..... | ........ | ...... | .... | ........ | ...... | ..... | ....... | do. | 45 |
| 152 6 | 15 3 | 8 4 | 5 0 | 26.4 | 2,200 | 3 | Various | 575 R<br>1,800 T | ..... | 140 | do. | 46 |
| 125 | 14 | ..... | ..... | 26.5 | 2,200 | 3 | 1 | ...... | 250 | 95 | ....... | 47 |
| 125 | 14 | ..... | ..... | 26.0 | 2,200 | .... | ........ | ...... | 250 | 95 | ....... | 48 |
| 200 | 23 | ..... | 8 0 | 28.3 | 6,000 | .... | ........ | 865 | ..... | 350 | Various | 49 |
| 140 | 17 | ..... | 7 0 | 18.0 | 1,800 | 2 | 1 | 650 | 250 | ....... | 4 6 | 50 |

44. Rateau Turbines. See Trans. I. N. A., 1904.
45. Do.
46. Do.
48. Brequet turbines. 49. Astern speed 16.7 knots. 50. Curtis turbines.

NOTE.—Also projected two vessels for Great Central Railway Co., two for Allan Steamship Co., two for the Metropolitan Steamship Co. (New York and Boston Service), and various foreign warships.

5. Smaller and more deeply immersed propellers, decreasing the tendency of the machinery to race in rough weather.

6. Lower center of gravity of the machinery as a whole, and increased headroom above the machinery.

7. According to recent reports, decreased first cost of machinery.

8. The adaptability of the turbine for greater power development in a single unit.

The application of the turbine to driving screw-propellers has presented a number of new problems to designers, such as

have been solved and reduced to more or less nearly standard practice in the case of the reciprocating-engine. Among these, the greatest importance attaches to the questions of *reversibility of turbines, efficiency of propellers,* and *economy at slow speeds.*

The problem of reversing has been met by the use of special reversing turbine drums, rotating idly in the exhaust-passage and upon the shafts of the low-pressure turbines when the ship is going ahead, but reversing the direction of rotation of the shafting and propellers when live steam is made to act upon the blades of the reversing-drums.

The determination of propeller proportions suitable for high speeds of rotation is still the subject of extensive investigation, although very satisfactory progress has already been made. The problem is to determine the proper diameter, amount and distribution of blade area, and the proper slip and pitch ratios to be used with the comparatively high rate of revolution of the steam-turbine.

High peripheral velocity of turbine blades may be obtained either by

(*a*) High rate of revolution and small diameter, or

(*b*) Large diameter and relatively low rate of revolution.

For satisfactory efficiency of propulsion with screw propellers, certain areas of propeller-blade surface are required, according to the thrust demanded, and it has been found advisable to limit the number of propellers to one upon each shaft. The shafts may be from one to four in number. There are three in the Carmania, and four in the two large Cunarders at present under construction. The requirement for a certain amount of area of blade surface with a limited number of propellers causes a limitation of the speed with which it is safe, or otherwise advisable, to rotate the shafts. This leads, in vessels of large displacement and high power, to the use of large diameters of the rotating members within the turbine casings, because otherwise the speed of rotation of the propellers would often be such as to cause low propulsive efficiency. The problem presents itself to the designer not as a propeller prob-

lem alone, capable of solution for any rate of revolution that may be adopted for the turbines, but as a question of the proper interrelation of steam velocities, diameter and rate of rotation of turbines, and size and proportions of the screw-propellers.

This suggests the chief difference in point of design between turbines for driving alternating-current machinery and those for rotating the shafts of screw-propellers. The stationary turbine may be operated at a high rate of revolution, with increasing efficiency and decreased size and weight of part accompanying the increase in speed. The marine turbine, especially for large powers, is called upon to turn the propellers at the relatively low rate of revolution giving satisfactory propulsive efficiency. Since both types require certain peripheral velocities in order to utilize the energy of the steam efficiently, the result is relatively high speed of rotation for the stationary turbine, with as small diameters as possible so as to reduce centrifugal forces; and large diameters of the marine turbine, with correspondingly low rates of revolution, for obtaining efficiency of screw-propellers.

Further difference in the arrangement of the two types is occasioned by the demand for close regulation of speed in the stationary turbine, and for reversibility in the marine turbine. The latter must be capable of sudden reversal of direction of rotation, and of ready handling at all speeds for maneuvering the vessel.

In general, with the larger turbine-boats that have been built, the economy has been somewhat lower at speeds below 14 knots than in boats driven by reciprocating-engines, but above this speed the turbine-boats have exceeded in economy, and the rate of increase with increased speed has been very marked. This is shown by the economy curves on pages 302 and 303 representing trials of torpedo-boat destroyers and cruisers.*

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* The curves and the table on pages 296–299 are from a paper by Mr. E. M. Speakman, Trans. Am Soc. Naval Architects and Marine Engineers, Vol. 13, 1905: "Marine Turbine Development and Design."

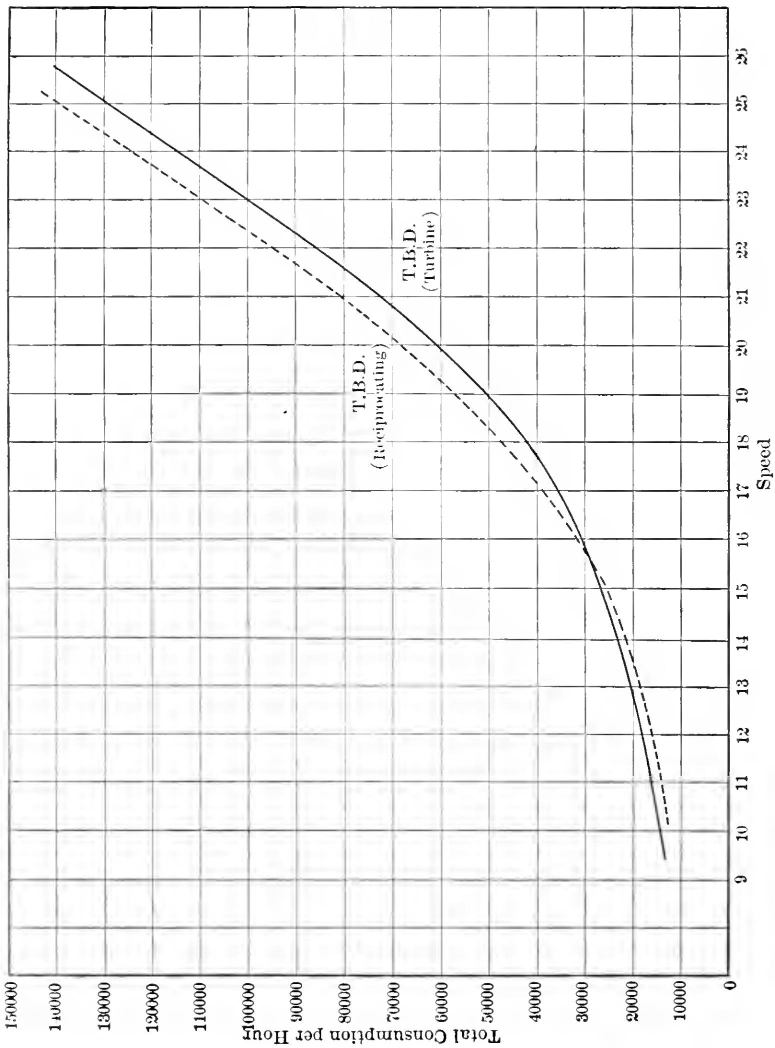



FIG. 120.—Economy curves of torpedo-boat destroyers.

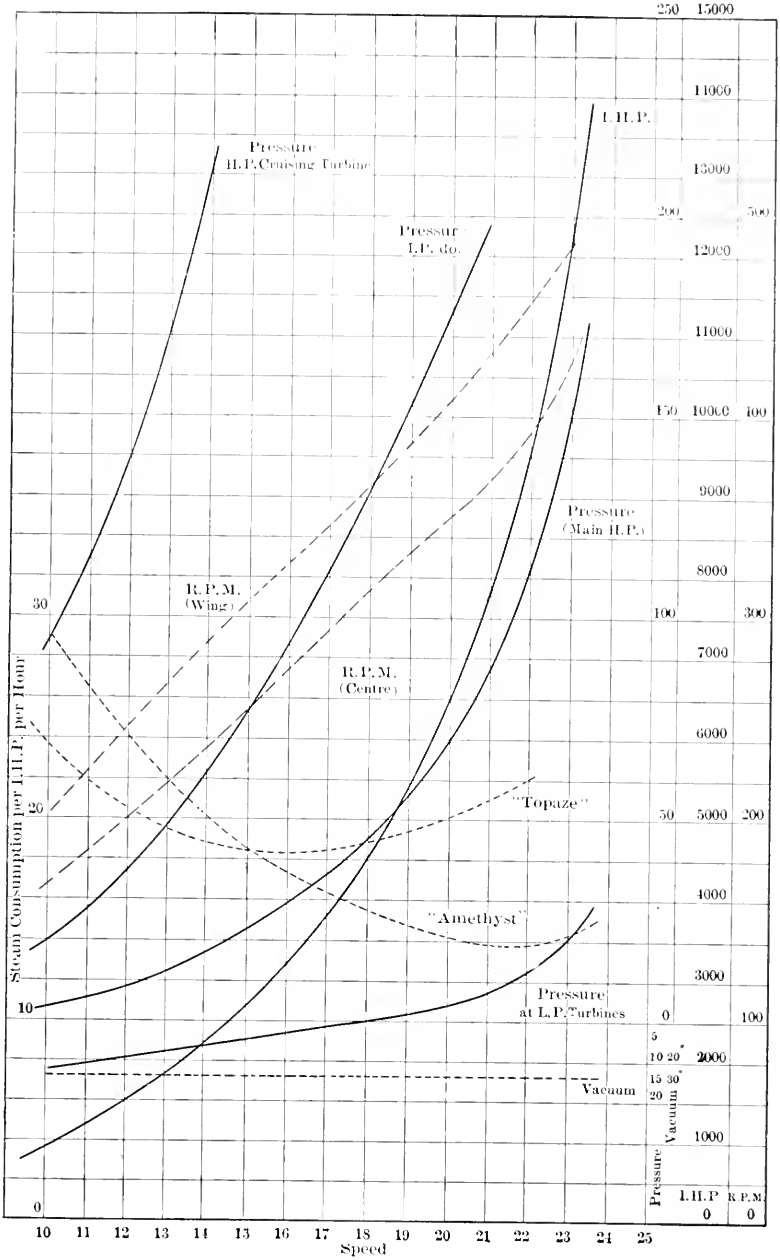



FIG. 121.—Economy curves of cruisers.

The following table * shows the steam consumption of the four Midland Railway steamers recently built and tested. The Antrim and Donegal are equipped with reciprocating-engines, each vessel having two sets of four-cylinder triple-expansion engines, each driving a three-bladed propeller. The cylinders are 23 inches, 36 inches, and two of 42 inches diameter, with 30-inch stroke of pistons.

| Speed in Knots per Hour. | Gallons of Water Consumed per Hour. | | |
|---|---|---|---|
| | Reciprocating, Antrim and Donegal. | Turbine. | |
| | | Londonderry. | Manxman. |
| 14 | 4,500 | 4,500 | 4,500 |
| 17 | 6,700 | 6,100 | 5,800 |
| 20 | 9,700 | 8,900 | 8,300 |
| 22 | .... | 13,600 | 12,500 |
| 23 | .... | ..... | 17,300 |

The arrangement of the turbines in the Londonderry and Manxman differs only in detail, but the turbines in the Manxman are larger, as they were designed for 25 per cent more power than the Londonderry. There are three turbines in each vessel, one high-pressure and two low-pressure. The reversing-turbines work upon the low-pressure shafts, and rotate in vacuum when not in use. Each of the three turbines drives a three-bladed propeller.

The dimensions of the four vessels are alike, with the exception that the Manxman is of slightly greater beam than the others. The length on the water-line is 330 ft.; moulded breadth, 42 ft.;† moulded depth, 25 ft. 6 in.

The amount of water consumed was measured during the progressive trials by counting the strokes of the feed-pumps.

Mr. Parsons has made the following ‡ prediction as to the future of the steam-turbine for marine use: ". . . With the evidence at present before us, I think we are safe in predicting

* London Engineering, August 4, 1905.

† Excepting the Manxman, of 43 feet beam.

‡ Trans. Inst. Marine Eng., London, 1904–5.

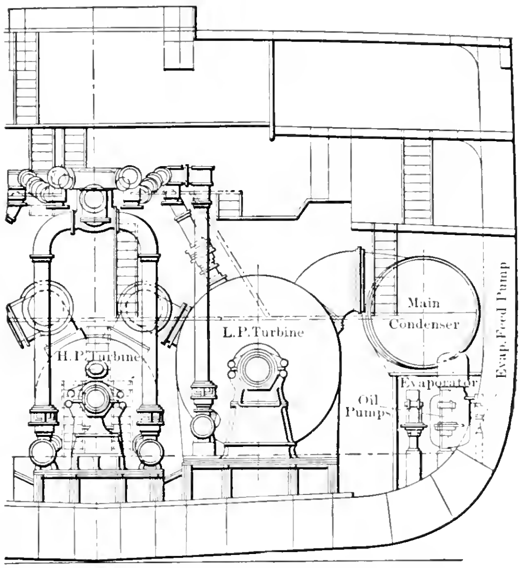



**Fig. 122.**—Cross-section through machinery space, steamship Carmania.

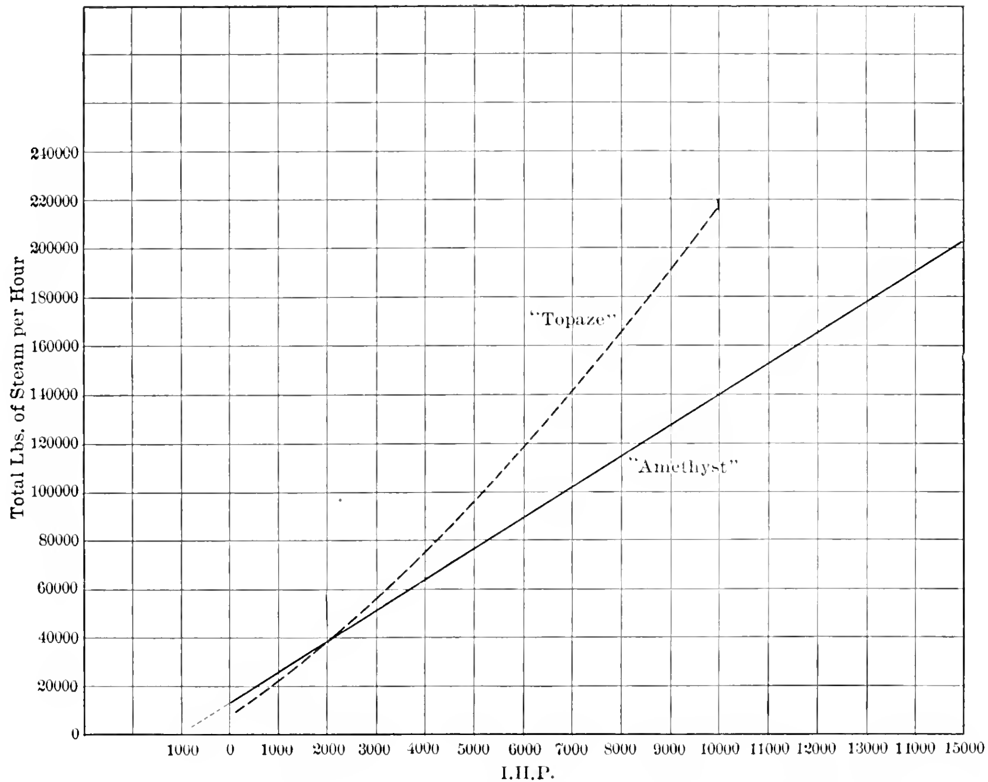



**Fig. 123.**–Economy of cruiser Topaze, with reciprocating-engines, compared with that of cruiser Amethyst, with turbines

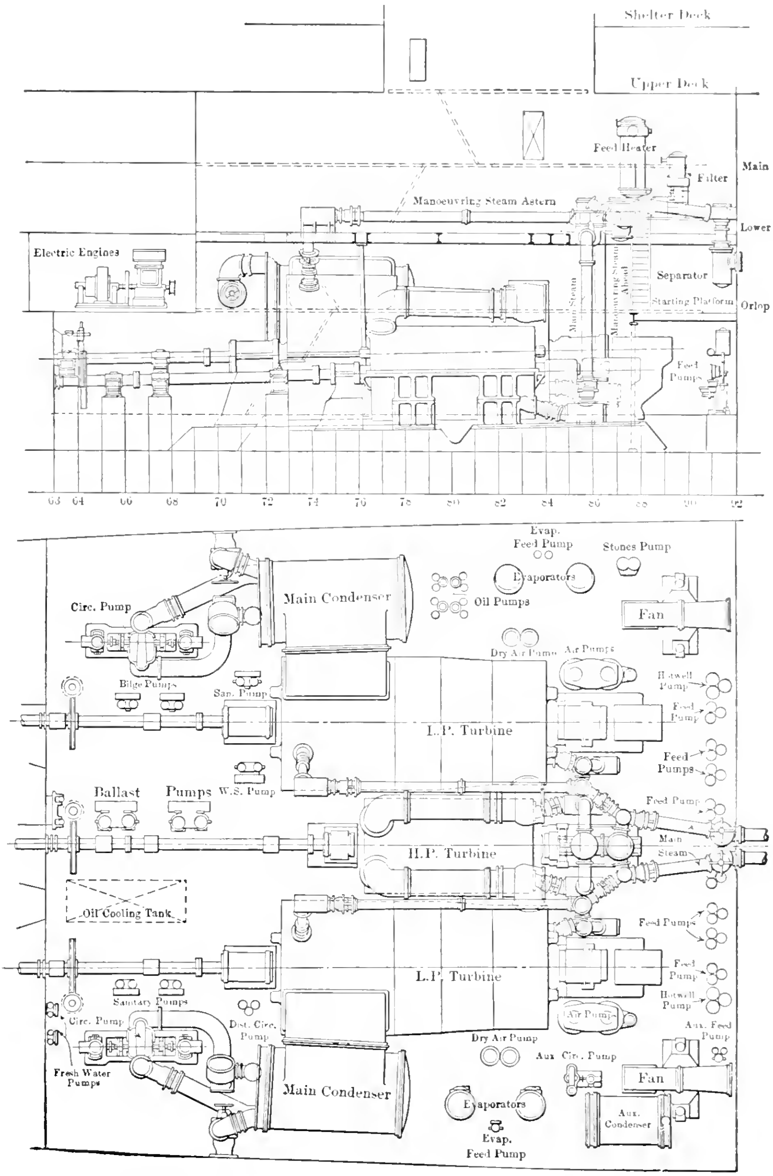



FIG. 124.—Arrangement of machinery in S.S. Carmania.
(From "Engineering," London, Dec. 1, 1905.)

that it will soon supersede entirely the reciprocating-engine in vessels of 16 knots sea-speed and upwards, and of over 5000 I.H.P., and probably also including vessels of speed down to 13 knots, of 20,000 tons and upwards, and possibly still slower vessels in course of time. At present it may, I think, be said that the above most suitable field comprises about one fifth of the total steam tonnage of the world; but it must be remembered that the speed of ships tends to increase, and the turbine to improve, and so the class of ships suitable for the turbine will increase."

The growth of the application of the Parsons turbine to steamship propulsion is represented in Fig. 125. At the left is shown the progress in application to war vessels, advancing from the experimental "Turbinia" to the battleship "Dreadnaught," and at the right the progress in application to merchant and passenger vessels, culminating in the production of the largest vessels afloat, the Cunard steamers "Lusitania" and "Mauretania," 785 feet long and of 25 knots speed. These remarkable vessels and their turbines are shown in Figs. 126, 127 and 128.

Fig. 129 shows a number of arragements of Parsons turbines suitable for various classes of vessels. It is to be noted that several of these arrangements show four propeller shafts. In general the requirement for great power in a ship calls for its distribution between several units as has been the case with the "Lusitania" and "Mauretania." It is possible and customary in certain classes of work to build single turbines to develop considerably more power than it is practicable to develop in a single reciprocating engine. But for very high powers the size of shafting and other parts becomes necessarily so great that it is often found advisable to distribute the power between several units, especially when the speed of rotation is low.

Figs. 130 to 134 inclusive show the steamer "Creole" and the turbines for propulsion. The ship was built by the Fore River Shipbuilding Company for the Southern Pacific Railway Company, is 440 feet long, and has a speed of about 17 knots. The small turbine shown in the foreground of Fig. 131 was de-

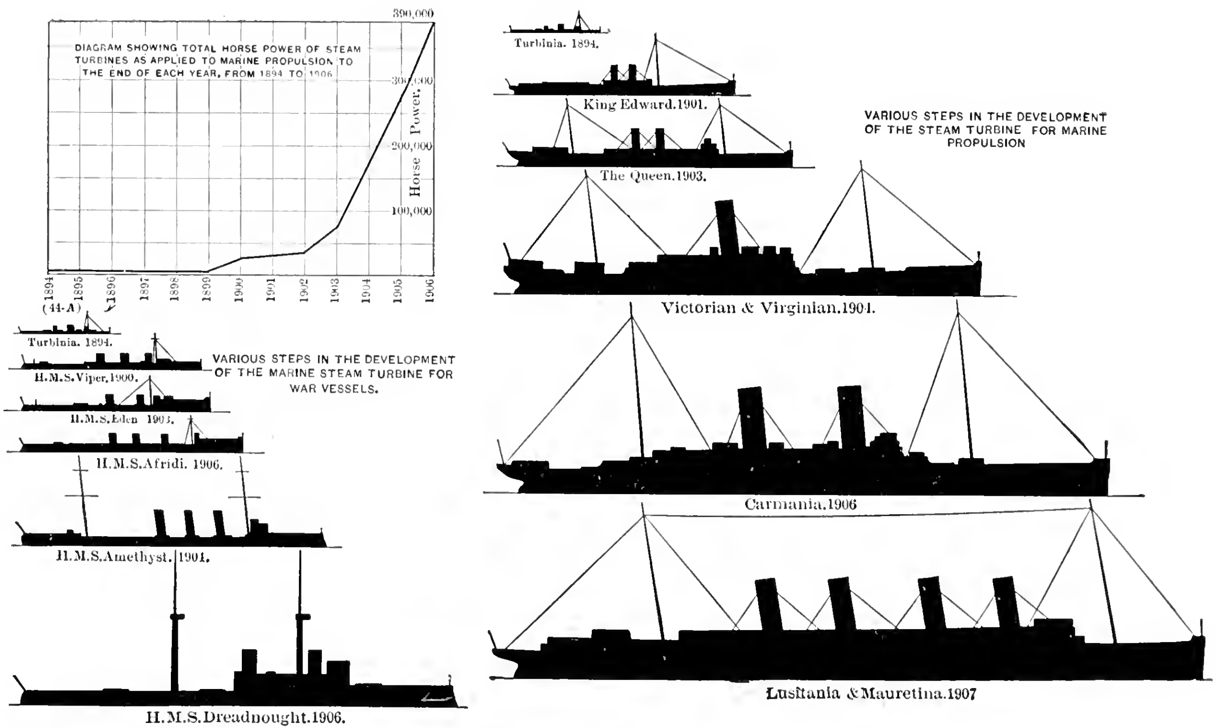



Fig. 125.—Showing the Successive Classes and Sizes of Vessels to which Parson's Turbines have been Applied. (Taken from illustrations for a paper by Mr. C. A. Parsons, read before the Inst. of Civil Engineers, London, June, 1907. Reproduced from "Engineering," London.

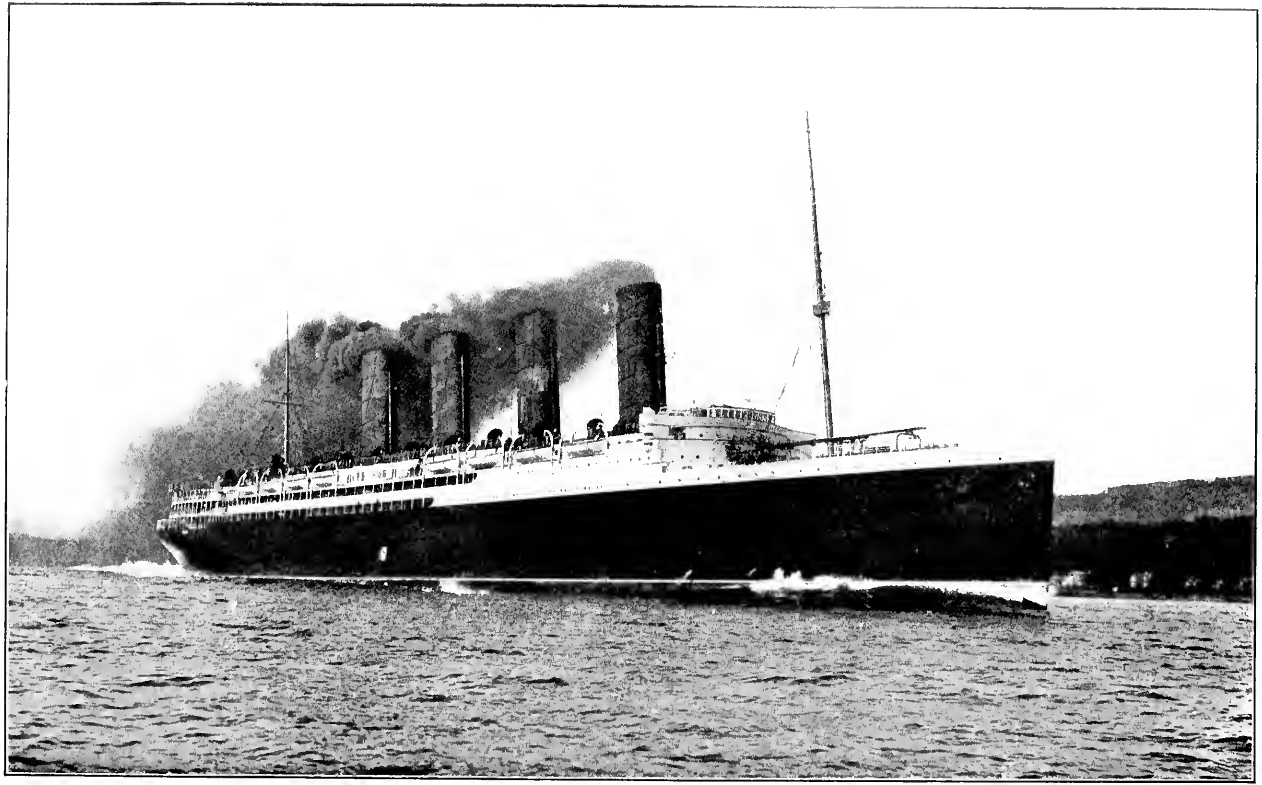


FIG. 126.—Turbine-steamer "Lusitania," Cunard Line. Length over all 785 feet. Breadth moulded 88 feet. Draught 33 feet 6 inches. Displacement 38,000 tons. Parsons' Steam-turbines, 68,000 Horse-power. Revolutions 160 to 180 per minute. 25 Cylindrical Boilers. 192 Furnaces. Steam-pressure 195 Pounds gage. Heating-surface 158,350 square feet. Grate-surface 4048 square feet. Howden Draught. Speed of Ship 25 knots. (From "Engineering," London.)

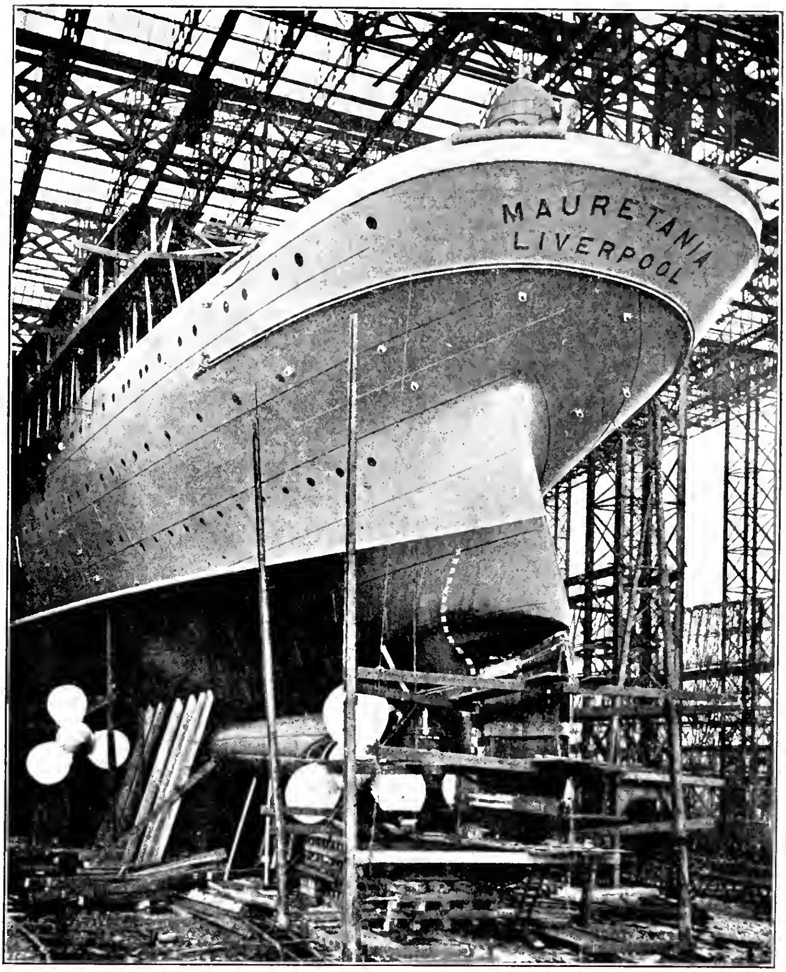



FIG. 127.—Stern View Showing Rudder and Two of the Four Propellers, Str. Mauretania, Cunard Line. Sister Ship to Lusitania. (From "Engineering," London.)

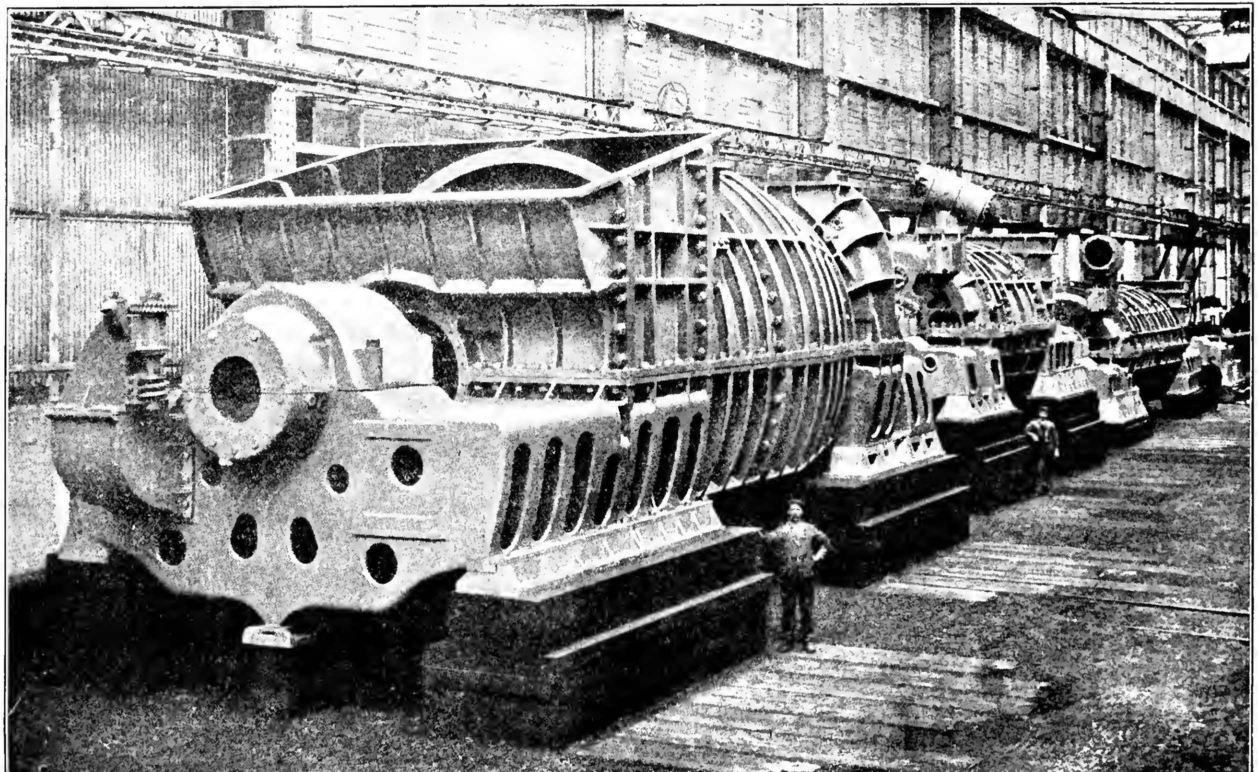
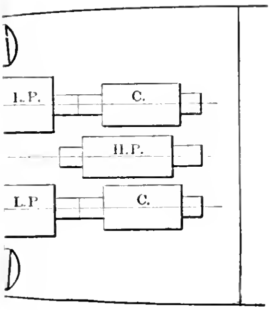
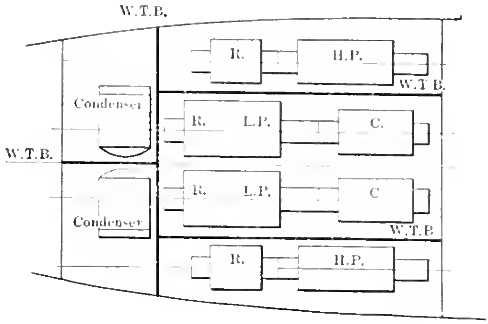


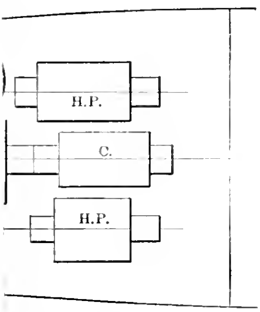
Fig. 128.—Turbines of the "Mauretania." View of One Astern Turbine, One L. P. Ahead, and One H. P. Ahead Turbine. (From "Engineering," London.)



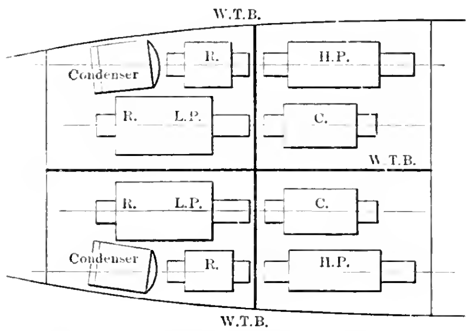

2 Reversing, 2 Cruising
1 Compartment.



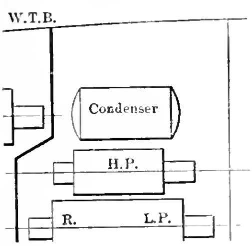

Warship, 4 Shafts, all Reversing. 5 Compartments, 2 Cruising Turbines.

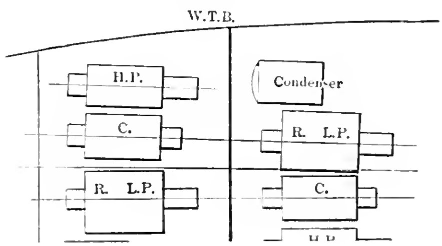


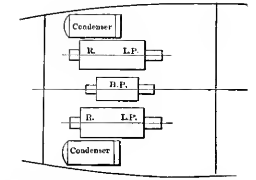

Shafts, 1 Reversing, 1
ne. 1 Compartment



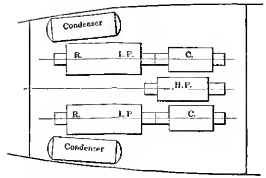

Warship, 4 Shafts, all Reversing. 4 Compartments, 2 Cruising Turbines.



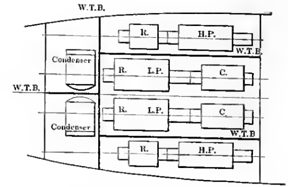





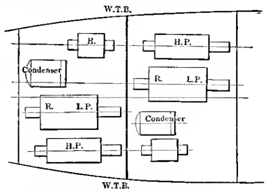

Merchant Ship, Standard Arrangement, 3 Shafts, 2 Capable of Reversing. 1 Compartment.



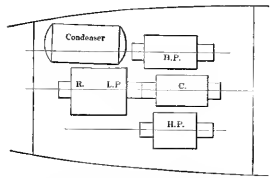

Warship, 3 Shafts, 2 Reversing, 2 Cruising Turbiues. 1 Compartment.



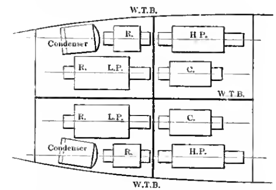

Warship, 4 Shafts, all Reversing 5 Compartments, 2 Cruising Turbines.



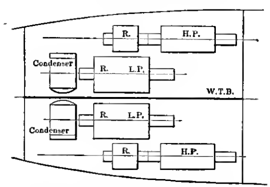

Merchant Ship, 4 Shafts, all Reversing. 2 Compartments.



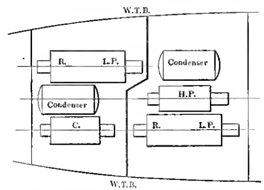

Torpedo Boat, 3 Shafts, 1 Reversing, 1 Crusing Turbine. 1 Compartment



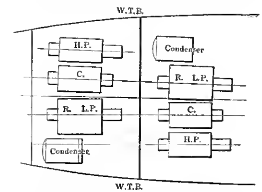

Warship, 4 Shafts, all Reversing. 4 Compartments, 2 Cruising Turbines.



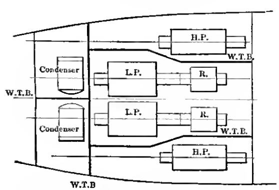

Merchant Ship, 4 Shafts, all Reversing. 2 Compartments.



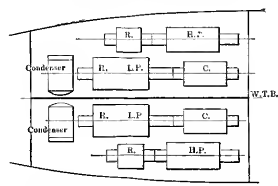

Warship, 3 Shafts, 2 Reversing. 2 Compartments, 1 Cruising Turbine.




Warship, 4 shafts, 2 Reversing. 2 Compartments, 2 Cruising Turbines.




Merchant Ship, 4 Shafts, 2 Reversing. 5 Compartments.




Warship, 4 Shafts, all Reversing. 2 Compartments, 2 Cruising Turbines.

FIG. 129.—Possible Arrangement of Parsons Turbines in Vessels of Various Classes. From a Paper by Mr. Chas. A. Parsons. Reproduced from Reprint in "Engineering," London, July, 1907.

*To face page* 312.

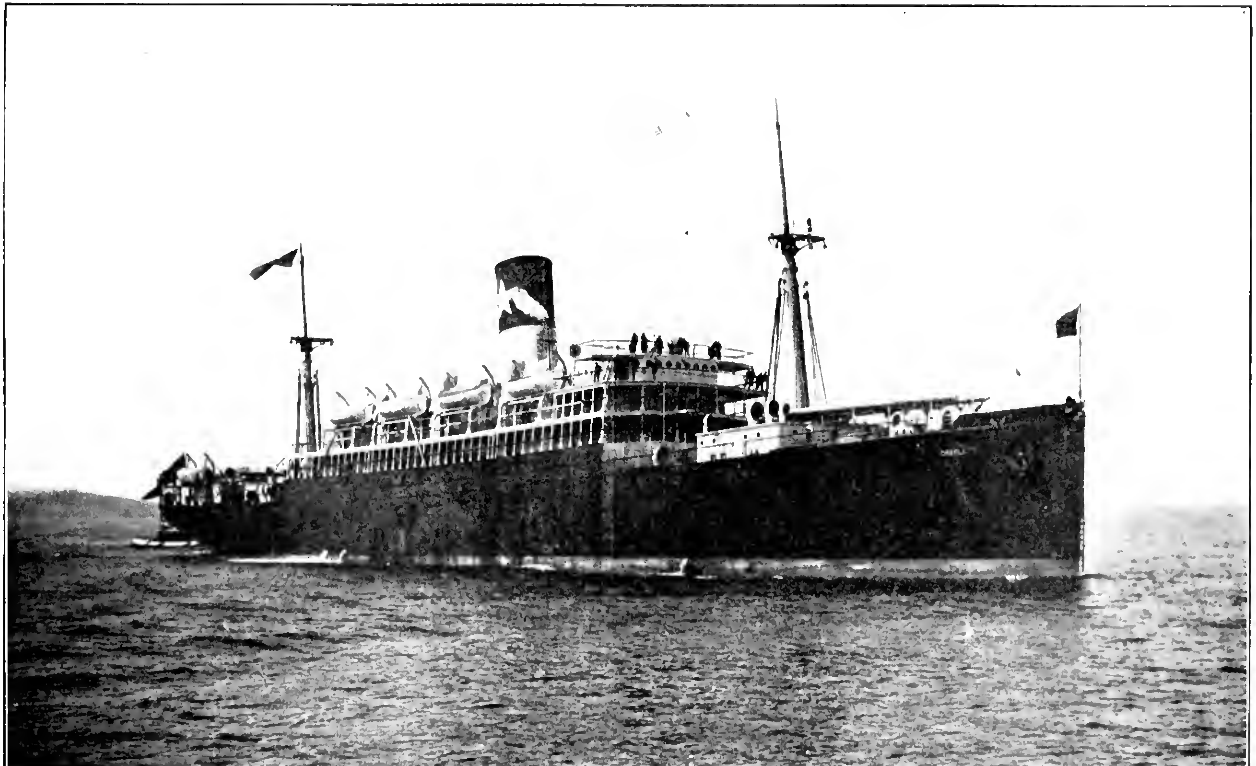


Fig. 130.—Steamer "Creole," built by Fore River Shipbuilding Company. 440 feet long. 8000 Horse-power. Curtis Turbines, 230 to 250 R.P.M. Speed of Vessel approximately 17 knots.

signed for a battleship tender, to develop 250 H.P. at 1200 r.p.m., while each of the main turbines for the "Creole" develops 4000 H.P. at about 235 r.p.m.

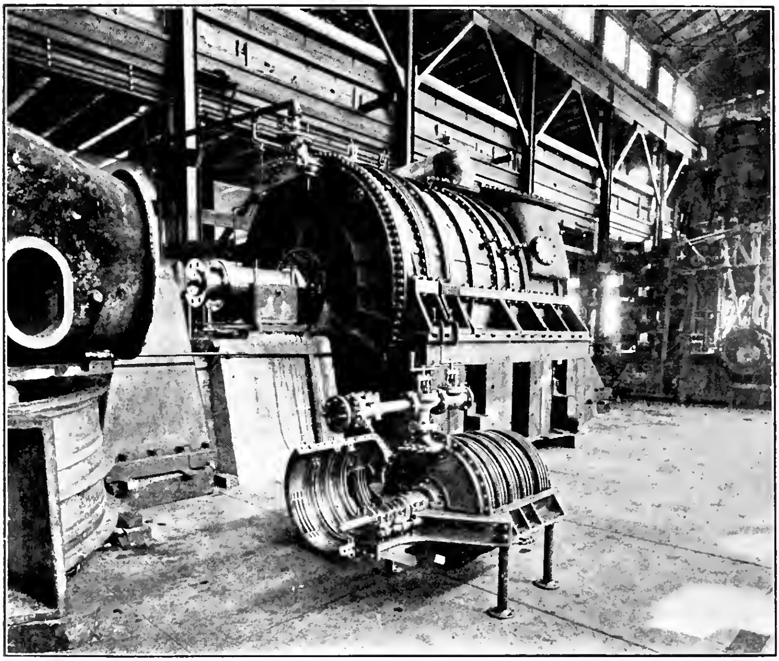


FIG. 131.—One of the two-4000 Horse-power Curtis Turbines of the Steamer "Creole," and a 250 Horse-power Curtis Turbine for Battleship Tender.

The first large turbine steamers to be put on the transatlantic service were the Allan line boats, "Victorian" and "Virginian," of about 18 knots speed. The arrangements of the turbines, condensers, shafting, and of the steam-piping, are shown in Figs. 135 and 136. One of the condensers of the "Victorian," with Mr. Parsons' *Vacuum Augmenter*, and with the air-pumps, is shown in Fig. 137.

With the development of the turbine has come the necessity for measuirng the power delivered to the shaft. Two methods are illustrated here, the first, that involving the application of a

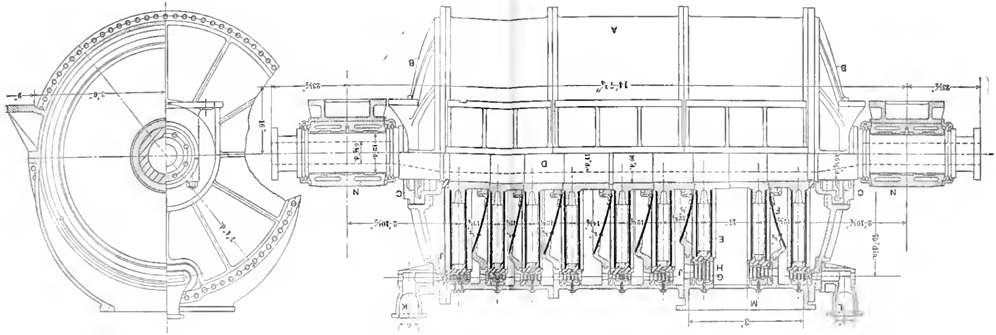


FIG. 134.—Curtis Marine Turbine as Fitted on Steamer "Creole."

*To face page* 315.

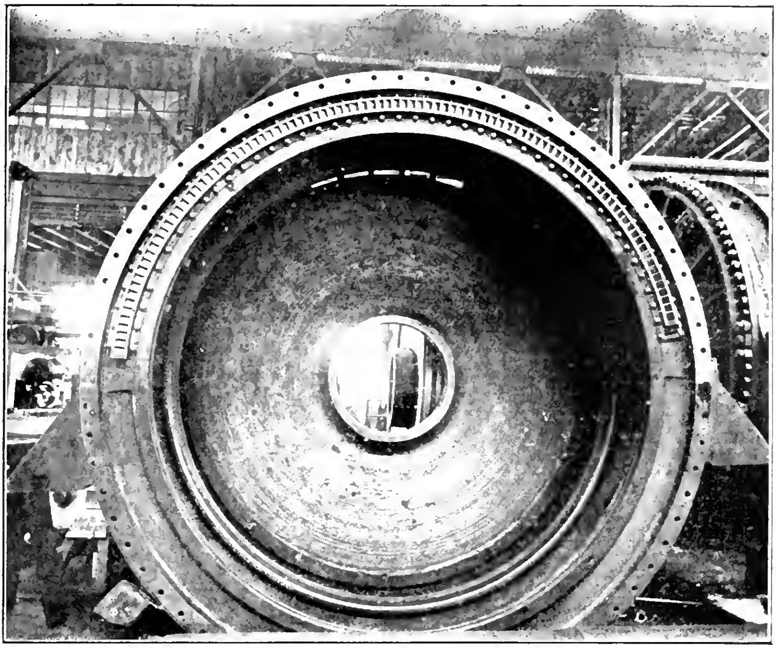


FIG. 132. Showing Nozzles of one of the Lowest Pressure Stages, Curtis Turbines for Steamer "Creole." The Turbines have Seven Stages Each.

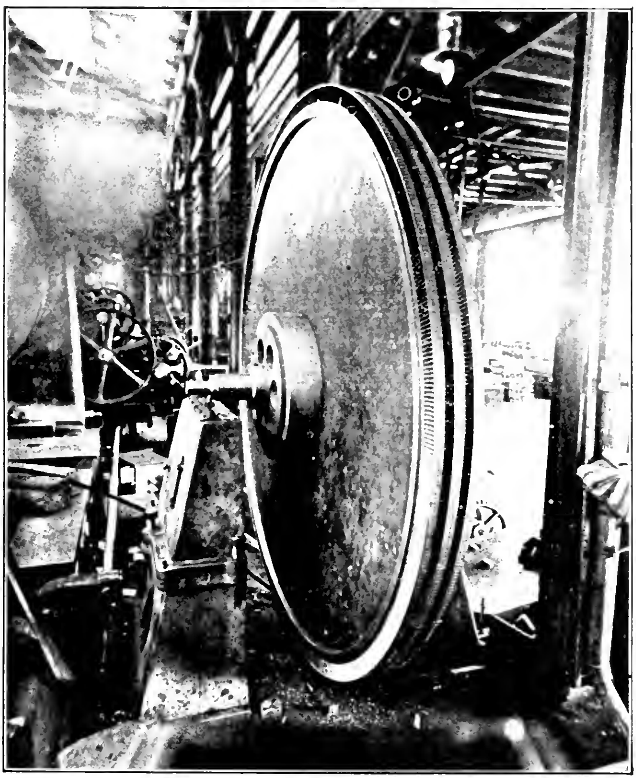


FIG. 133.—One of the Rotating Wheels, Curtis Turbines for Steamer "Creole."

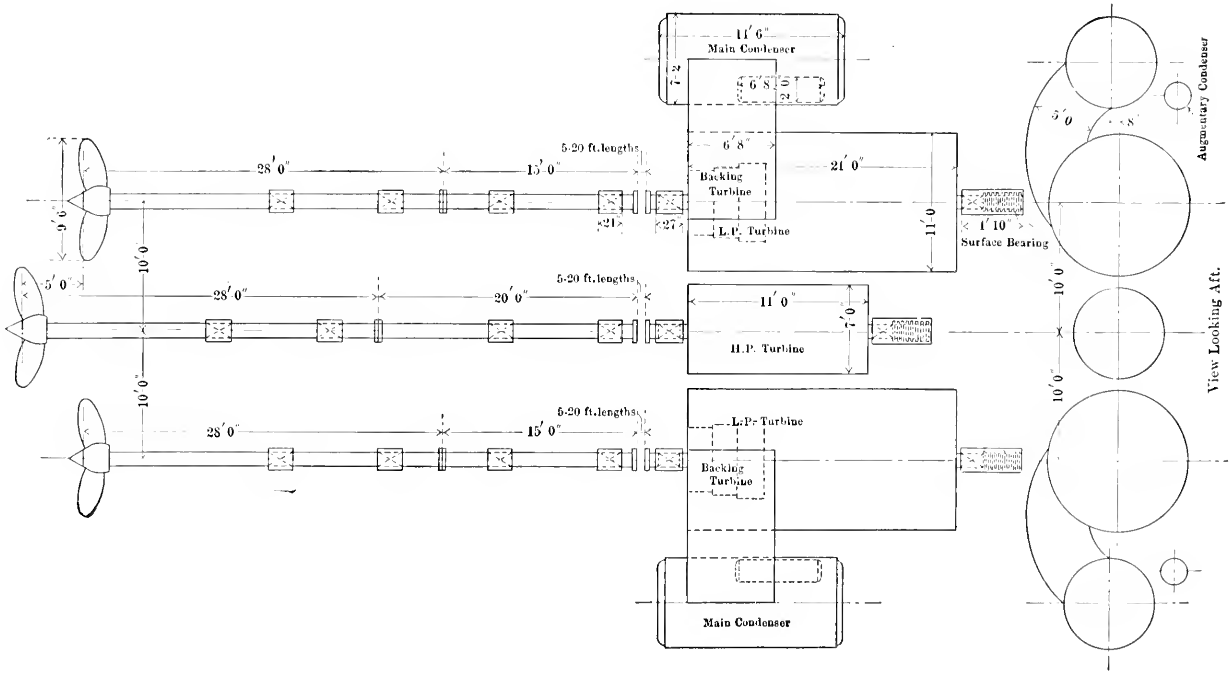



Fig. 136.—Arrangement of Turbine Shafting and Condensers. Allen Line Steamship "Victorian."

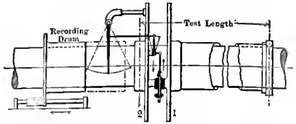



Fig. 138.—Shaft Fitted with Foettinger Torsion Meter.

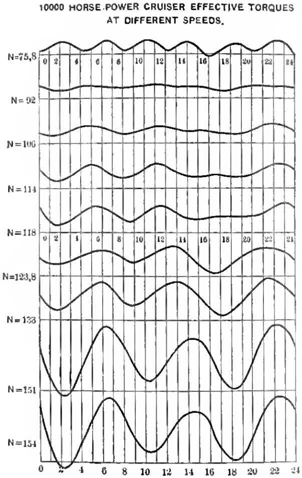



Fig. 139.

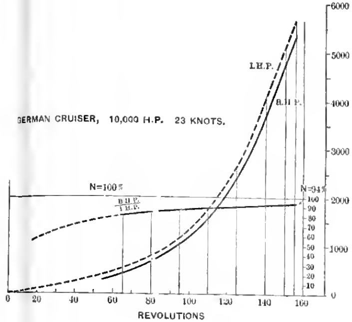



Fig. 140.

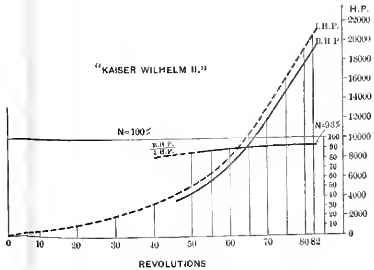



Fig. 141.

Diagrams obtained with Foettinger Torsion Indicator from Reciprocating Engines, and showing the Mechanical Efficiency or Relation between the Delivered and Indicated Horse-power of the Engines

*To face page* 317

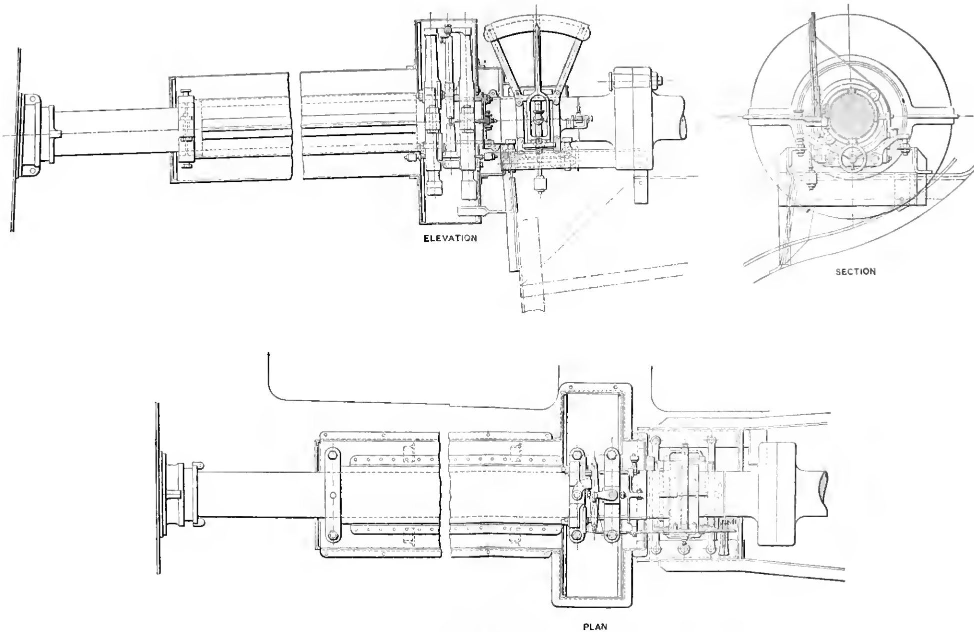



Fig. 140.—Foettinger Torsion-meter applied to German Cruisers. Reproduced from Paper by Mr E. M. Speakman, Inst. of Engineers and Shipbuilders of Scotland. Vol. L.

*To face page* 317.

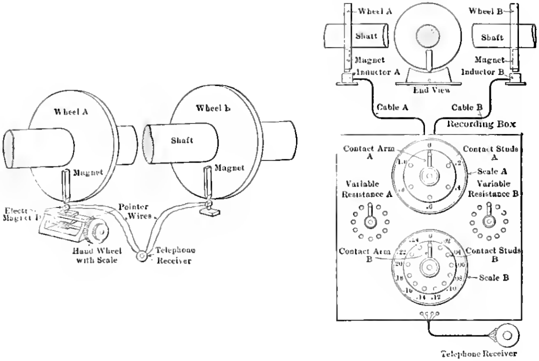



FIG 142 Diagrammatic Representation of Denny and Johnson Torsion-meter.

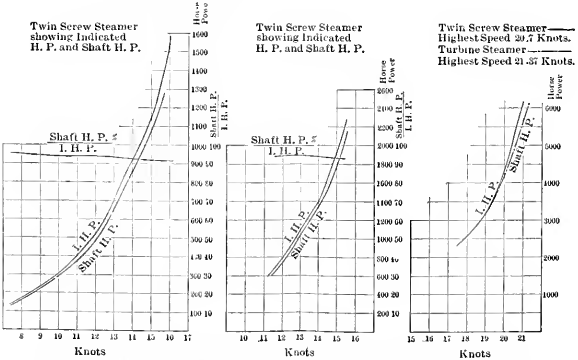



FIG. 143.—Diagram showing Relation between Delivered and Indicated Horse-power. Reproduced from "Engineering," London.

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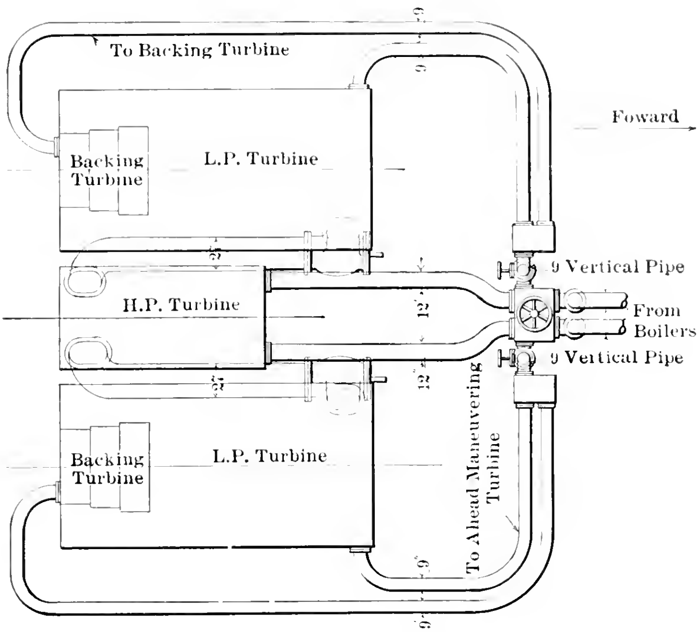


Fig. 135.—Arrangement of Steam-piping, Steamer "Victorian."

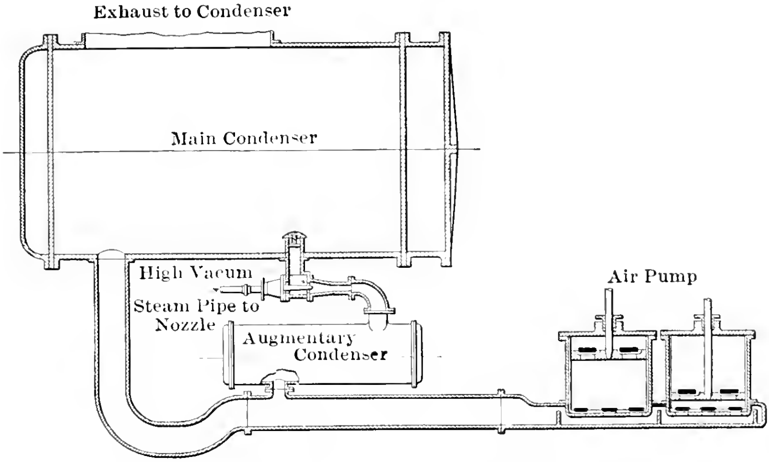


Fig. 137.—Condensers and Air-pumps, Steamer Victorian, Allan Line.

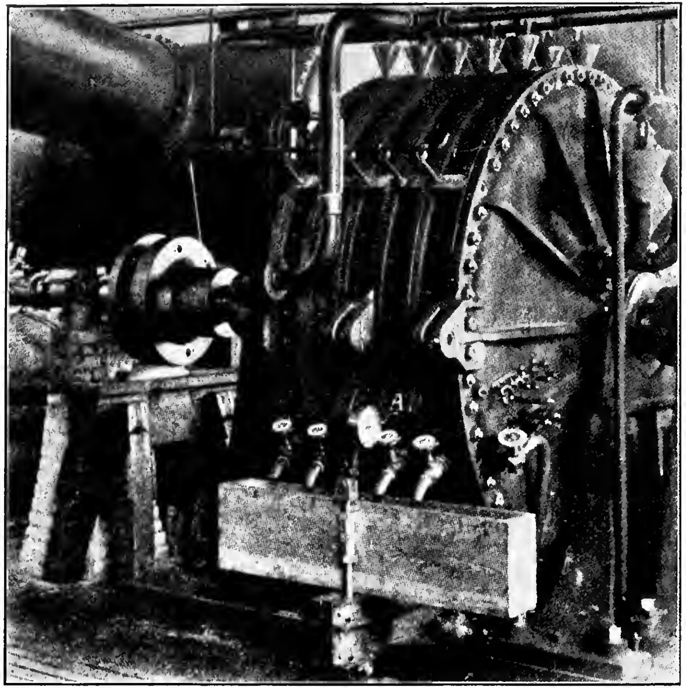


Fig. 144.—Exterior of Water-brake.

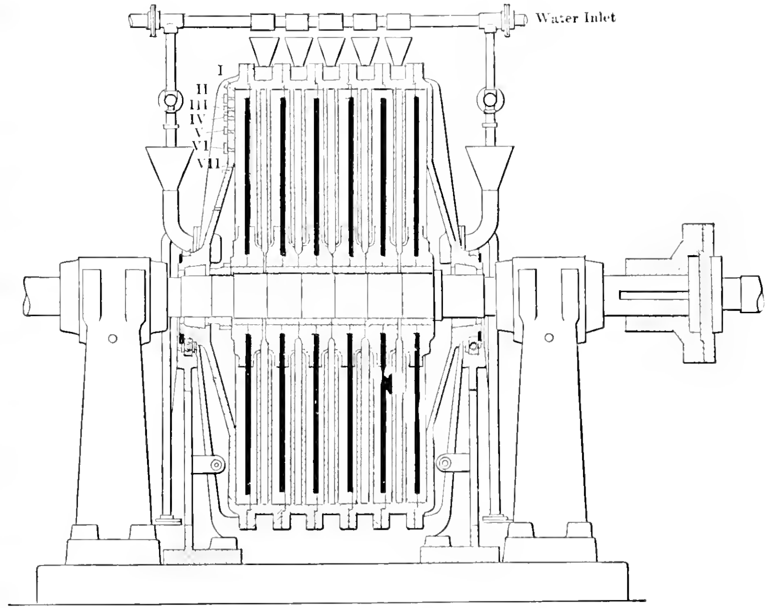



Fig. 145.—Water-brake for Absorbing the Power of a 3000 Horse-power. Curtis Marine Turbine.



Reproduced from the Zeitschrift des Vereins d. Ingenieure.

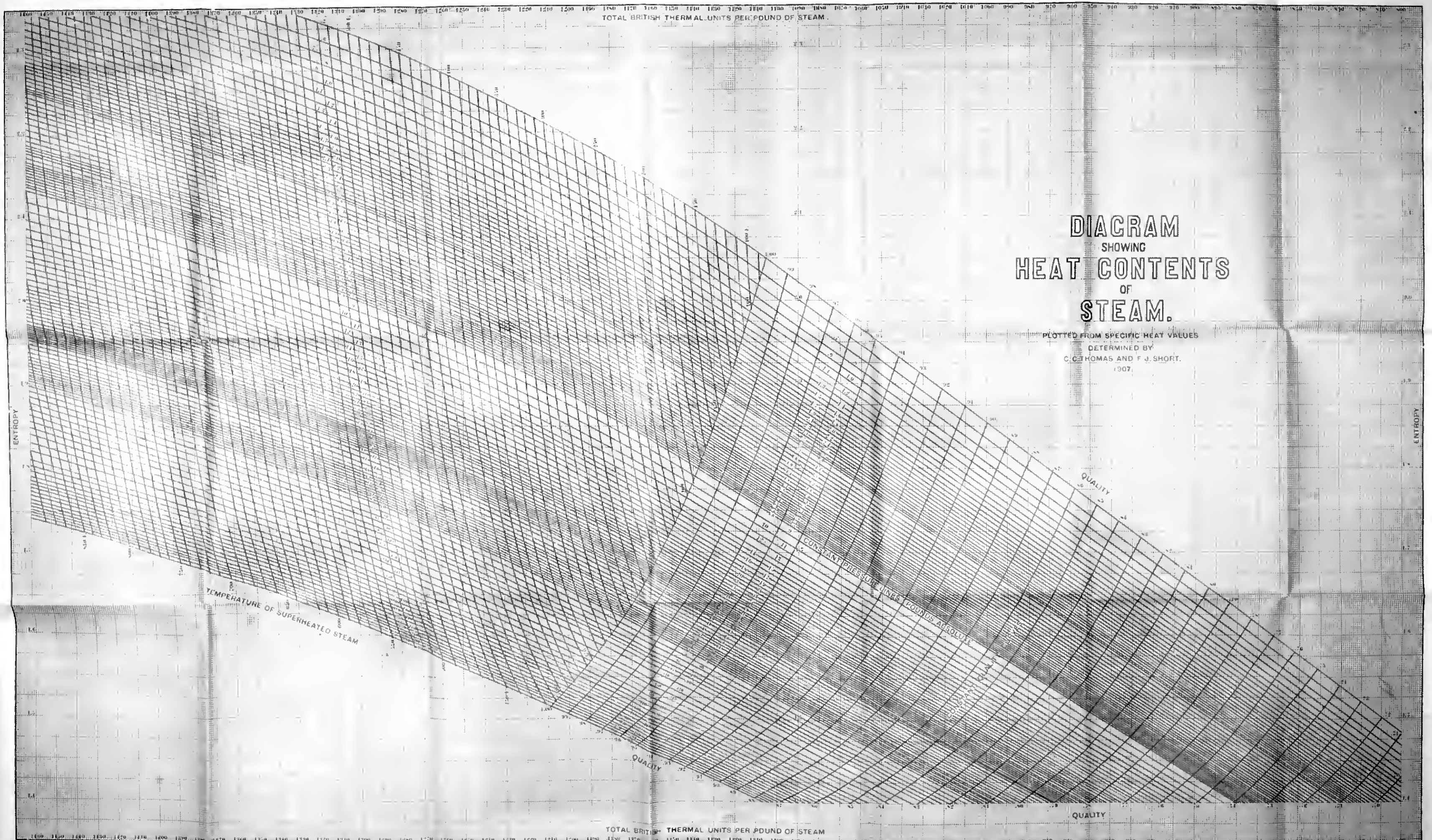
water-brake to the shaft; and the second, that in which the arc of torsion of a certain length of shafting is measured while power is being transmitted by the shaft. All shafts twist to some extent under the influence of torque, and for stresses below the elastic limit of the material the arc of torsion is proportional to the torque. The first successful torsion-meter for turbine use was developed in Germany by Dr. Foettinger, of Stettin, upon the basis of the extensive experiments made by Hermann Frahm of Hamburg to ascertain the extent of the torsional vibration of the shafting of reciprocating engines. The Foettinger torsion-meter is shown in Fig. 138, and diagrams obtained by its use in Figs. 139, 140, 141, inclusive. The Denny-Johnson torsion-meter was developed by Messrs. Denny Brothers of Dumbarton, Scotland. This meter is represented in Fig. 142, and results obtained are shown in Fig. 143. Torsion-meters have yielded most valuable information as to the mechanical efficiency of reciprocating marine engines, and have thereby contributed materially to the available information concerning ship propulsion.

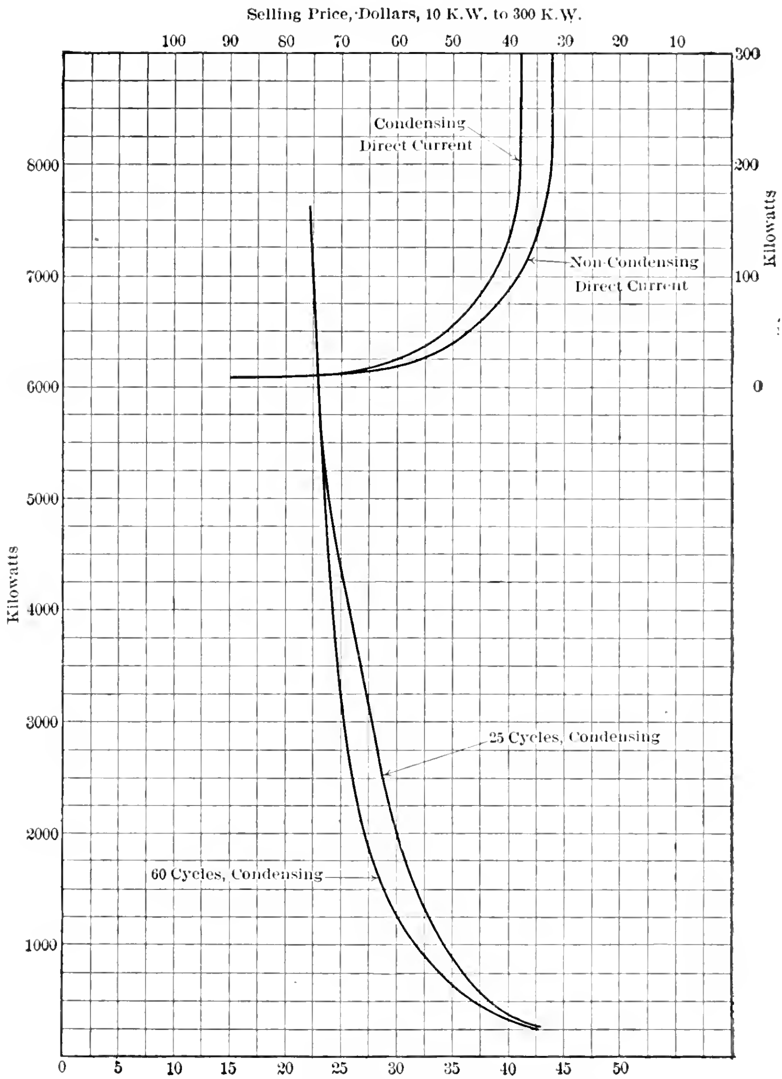
Water-brakes are not convenient for application aboard ship, but are extensively used in shop tests of turbines. Numerous forms of water-brake have been devised, but in all the power developed by the turbine is expended in setting water in motion by means of rotating metal discs or wheels. The torque is measured by weighing the pull on a brake-arm attached to the casing in which the rotating member is enclosed. The casing tends to rotate because of the action of the water, which is set in motion by the rotating discs of wheels. The water is of course heated by the frictional resistance opposed to its motion. Figs. 144 and 145 show one form of water-brake which is successfully used in turbine tests.

# APPENDIX.

**Cost of Steam-turbines.**—The curves on page 321 represent the selling price of turbines and generators combined, as quoted by various builders in 1905. Averages of quotations from the different firms were taken for plotting the curves. The upper right-hand section of the page gives curves of selling prices for comparatively small machines, connected to direct-current generators of from 10 K.W. to 300 K.W. capacity. The lower curves are for large machines—from 300 K.W. to 7500 K.W. —with alternating-current generators The curves are given to show the manner in which the selling price varies with the conditions of operation, but the values represented do not correspond exactly with the quotations of any one company.


A


TOTAL BRITISH THERMAL UNITS PER POUND OF STEAM.
DIAGRAM
SHOWING
HEAT CONTENTS
OF
STEAM.
PLOTTED FROM SPECIFIC HEAT VALUES
DETERMINED BY
C. C. THOMAS AND F. J. SHORT.
1907.
ENTROPY
ENTROPY
QUALITY
QUALITY
QUALITY
TEMPERATURE OF SUPERHEATED STEAM
CONSTANT PRESSURE LINES POUNDS ABSOLUTE
TOTAL BRITISH THERMAL UNITS PER POUND OF STEAM




Selling Price, Dollars.
For Machines from 300 K.W. to 7500 K.W. {Alternating Current

# EXAMPLES.

## SET NO. 1.

### TEXT REFERENCE, PAGES 6–20.

1. One quarter pound of steam flows per second from a vessel fitted with an orifice having a least cross-sectional area of .025 sq. in Let the specific volume of the steam while in the orifice be 2.0 cu. ft. per pound.

(*a*) Compute velocity of flow.

(*b*) Compute the reaction accompanying the flow.

2. If the steam should act upon the buckets of a turbine-wheel, leaving same at a velocity of 1000 ft. per sec.,

(*a*) What horse-power will be given up to the wheel, assuming there are no frictional losses?

(*b*) Compute the efficiency of wheel from the above considerations.

(*c*) If the exhaust, at 1000 ft. per sec., should act upon the buckets of another wheel, leaving same at 300 ft. per sec., how much power would the two wheels together deliver, disregarding losses?

(*d*) What would be the efficiency of the system?

3. A vane such as that shown in Fig. 9, page 18, moves with a velocity of 1200 ft. per sec., and is acted upon by a jet of steam having an initial velocity $V_1$ of 3400 ft. per sec. The angle $\alpha = 24$ degrees and $\beta = 30$ degrees.

(*a*) If one quarter pound steam per sec. acts upon the vane, compute the impulse of the jet upon the vane.

(*b*) Find the proper value of the angle of the entering side of the vane, so that the steam may enter without loss from impact.

## SET NO. 2.

### TEXT REFERENCE, CHAPTER III.

A pound of water at 520 degrees F. absolute is heated until its temperature becomes 790 degrees absolute.

(*a*) Assuming its mean specific heat to be 1.006 for the temperature range in question, how much heat is required to accomplish the rise in temperature?

(*b*) What increase in entropy has accompanied the addition of heat?

(*c*) If further heat be added until the entropy of the resulting mixture of steam and water is 1.55, as shown on the chart at the back of the book, what will be the percentages of steam and of water present?

(*d*) If the mixture should expand adiabatically in a nozzle to a pressure of 10 lbs. absolute, what would be the resulting velocity of flow from the nozzle under ideal conditions?

(*e*) What would be the quality of the exhaust from the nozzle?

(*f*) If sufficient heat had been added in (*c*) to evaporate the pound of water into dry and saturated steam, how much *more* heat would be required to superheat the dry steam to a temperature 100 degrees above the saturation-point, assuming the mean specific heat of superheated steam to be .58?

(*g*) To what temperature would the superheated steam have to fall, adiabatically, in order to become just dry and saturated?

(*h*) If the expansion indicated in (*g*), of the superheated steam, occurred in a suitable nozzle, so that the energy liberated all appeared as kinetic energy of flow, compute the velocity of the issuing steam-jet.

## SET No. 3.

### Text Reference, Chapters IV and V.

Design a nozzle for carrying out the expansion of .25 pound steam per second, under the following conditions:

Let the initial pressure be 165 pounds absolute $=p_1$.

" " final " " 2 " " $=p_2$.

" " loss of energy in the passageway be that corresponding to $y=.14$.

Let the steam before entering the nozzle be 98.5 per cent dry.

Find the proper cross-sectional areas for the nozzle at points where the pressure is 95 pds., 75 pds., 60 pds., 45 pds, 30 pds., 15 pds., and 2 pds., absolute, per sq. in.

Let the interior of the nozzle be conical in form, 4″ long.

Make a sketch of the nozzle to scale, and plot curves of pressure fall and velocity similar to those on page 149.

Typical calculations are given on page 85.

## SET NO. 4.

### Text Reference, Pages 151–158.

Let steam expand in the nozzles of a simple impulse turbine (de Laval type) from 120 pounds absolute to a vacuum of 27″ mercury. Let the nozzle make an angle $\alpha=28°$ with the plane of rotation of the buckets. Let the peripheral velocity of the buckets be 1300 feet per second. Find steam velocity from Plate XI.

(*a*) Draw velocity diagrams, allowing for no losses, and compute the energy given up to the buckets, per pound of steam, and compute the steam consumption of the deal turbine, and the efficiency. Make a sketch of the buckets on the velocity diagram.

(*b*) Let the loss of energy in the nozzles correspond to $y=.15$, and in the buckets let $y'=.13$.

(1) Draw velocity diagrams, and sketch in the bucket outline. Note the change in bucket angles, made necessary by the losses.

(2) Compute the work done per pound of steam, and the steam consumption per horse-power hour.

(3) Compute the efficiency of the turbine.

(4) If the revolutions of the wheel are 14,000 per minute, find diameter of mean bucket circle.

(5) If seven nozzles are used at ma imum load of 75 K.W., find least diameter of the nozzles, by means of the curve of discharge on Plate XI.

## SET. NO. 5.

### Text Reference, Pages 158–175.

(*a*) Draw velocity diagrams for an impulse turl ine of two stages and three rows of moving blades in each stage, according to the following data.

Let the turbine be required to develop 1000 K.W. at full load and 1400 K.W. at maximum overload. Efficiency of generator=94%. Let the initial pressure at inlet be 145 pds. gauge, and let the steam expand to 15 pds. abs. in the first nozzles, and in the second nozzles from 15 pds. abs. to a vacuum of $28\frac{1}{2}$ in. mercury. Let the angle of nozzles with plane of rotation of buckets be 22°. Let peripheral velocity of buckets be 420 ft. per sec. Assume that the frictional losses are represented by the values of $y$ given on pages 164 and 168 respectively, and let the work lost because of journal friction, windage, and leakage be 25% of the work done by the steam. Draw diagrams as on Plate XII, and compute steam consumption per H.P. as on pages 166 and 169, arranging for the maximum overload requirement.

Let R.P.M. be 1800. Compute height of second stage nozzles following the method given on pages 170, etc., and according to the following data: Let thickness of nozzle walls be 0.075 in. and let pitch of nozzles be 1.5 in. Let the nozzles subtend an angle at center of turbine shaft, of $\Delta=130$ deg. If height of first row of buckets is $2\frac{1}{2}\%$ greater than that of the nozzles, and if height ratio for second stage is 1.6, compute height of last buckets in the stage.

(*b*) A turbine takes steam at 175 pds. abs. and 125 deg. F. superheat, and expands adiabatically to a vacuum of 27.8 in. mercury. Find available energy from the Mollier Heat Diagram opposite p. 320. How much steam would a perfect engine use under these conditions? If a test of an actual

turbine shows a steam consumption of 12 pds. per horse-power hour, what is the efficiency of the engine? (See pages 175–6).

Let the horse-power be 3000 and let the R.P.M. = 900. Let the bucket speed be 350 ft. per sec. Compute diam. of turbine.

Let the turbine have six stages and let the energy distribution aimed at be, First stage, 0.25 E. and each succeeding stage 0.15 E.

Let the first stage efficiency be 0.45 and for each remaining stage let efficiency = 0.50.

Find the area required through nozzles of last stage, in order to provide for 3000 H.P. Assume the nozzle particulars to be the same as stated at bottom of page 187, and compute necessary height of nozzles for the last stage of the turbine.

Note that the diagram on the back cover of the book shows an expansion curve representing the calculated expansion of steam, as given on page 186.

The heat contents of steam may be taken from either the Heat Diagram opposite page 320 or from the one on the back cover, but the former is preferable, especially as it is well to become familiar with the Mollier Diagram as used in practice.

## SET NO. 6.

### Text Reference, Pages 189–195.

Turbine of the Parsons Type, 2500 B.H.P., 1800 R.P.M.

Initial steam pressure, 165 pds. abs.

Initial superheat, 100 deg. F.

Vacuum $28\frac{1}{2}$ inches mercury.

Ratio peripheral to steam velocity = 0.55.

Turbine to have 3 cylinders, and let the mean peripheral velocities in the cylinders be respectively 140, 220 and 325 ft. per sec.

Let the heat absorbed by the various cylinders be, H.P. 28%, I.P. 32%, and L.P. 40%.

Assume adiabatic expansion and calculate number of rows and mean diameters of the cylinders.

Let the annular space occupied by blades have 2.6 times the cross sectional area required for steam flow.

Let the steam consumption at full load be 13 pounds per B.H.P. hour. Assume that the steam pressure after passing the throttle valve is 145 pds. abs. and has dropped to this along a constant heat curve. Find specific volume at entrance to the first cylinder. Compute blade lengths at entrance to and at exit from each of the cylinders, as on pages 193–4. Let the steam velocity at the last rows of the L.P. cylinder be 1000 ft. per sec. instead of remaining constant during passage through that cylinder.

Tabulate results as on page 195.

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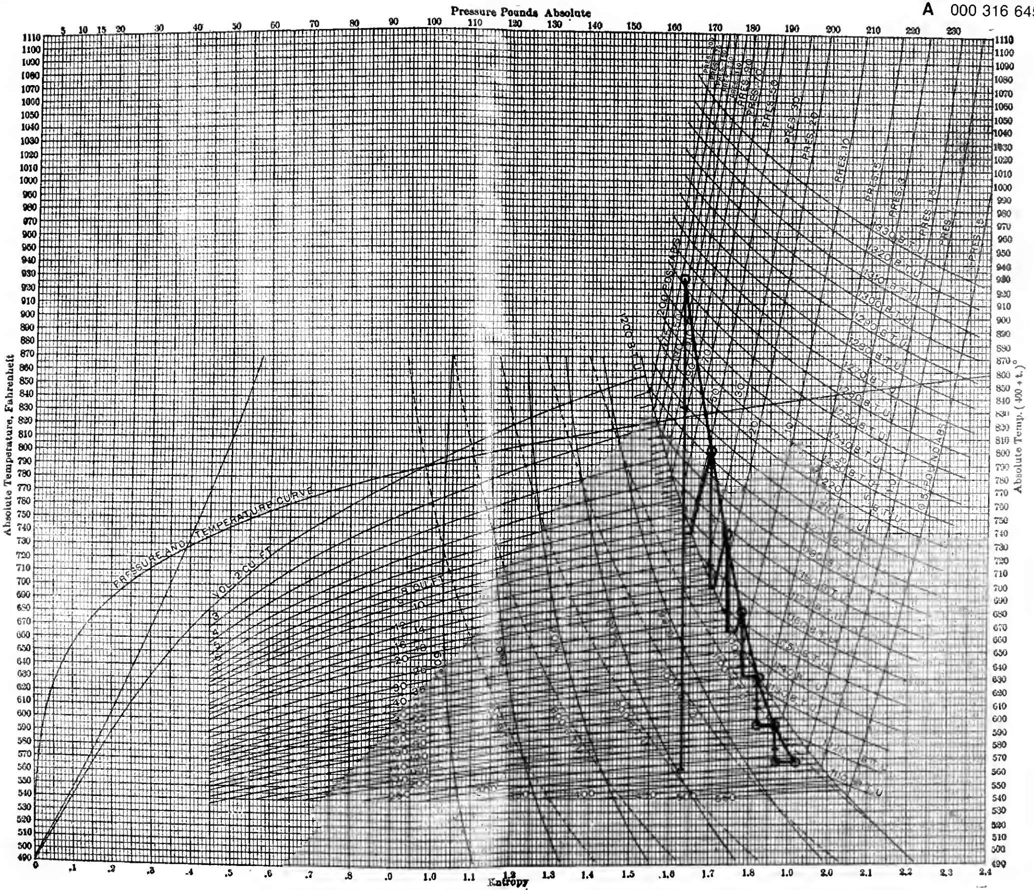
Pressure Pounds Absolute
Absolute Temperature, Fahrenheit
Absolute Temp. (460 + t.)
Entropy
PRESSURE AND TEMPERATURE CURVE
VOL. 2 CU. FT.
200 PDS. ABS.
10 LB. POUNDS ABS.